POWER

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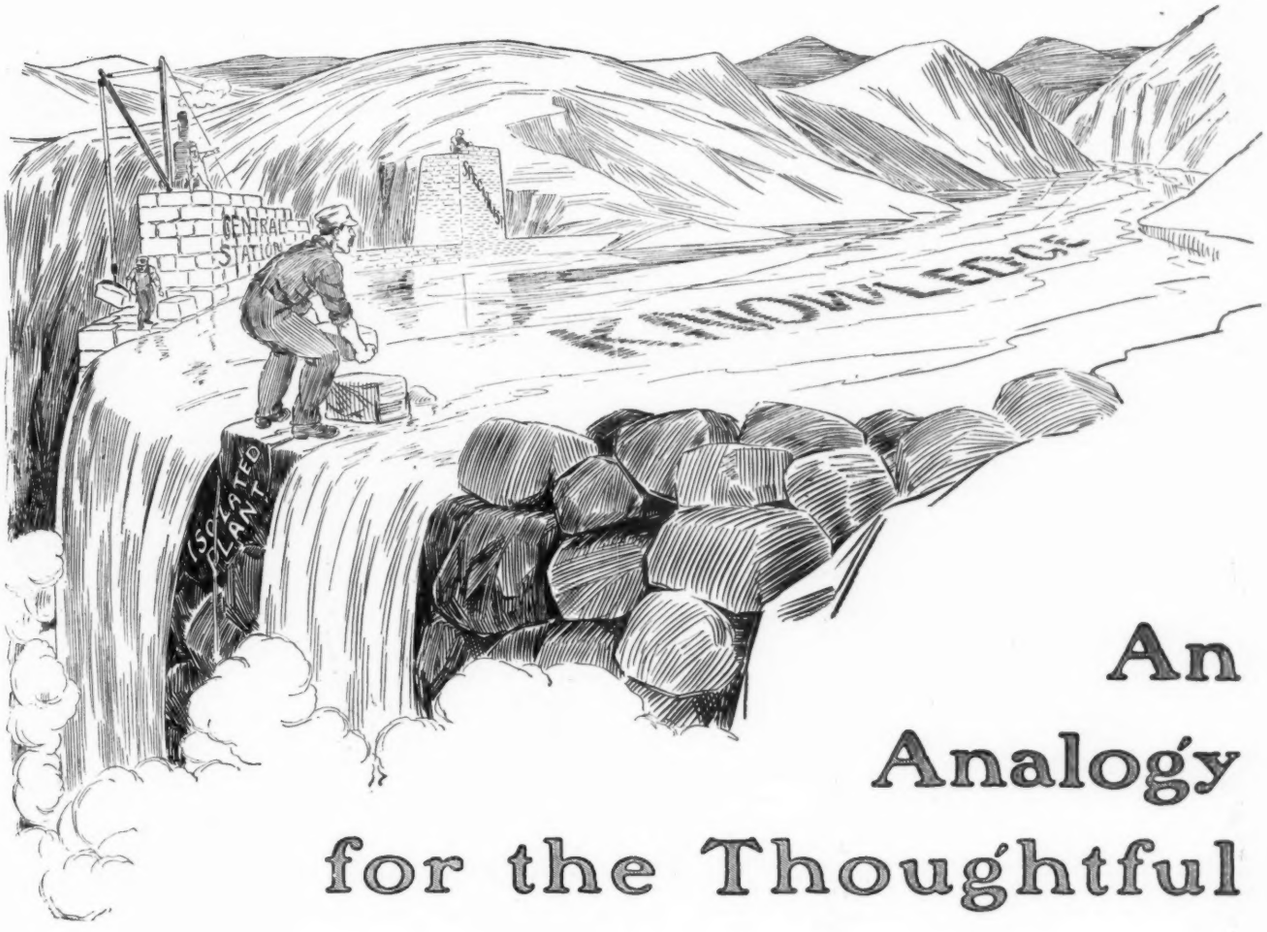


An Analogy for the Thoughtful

IN the Engineering Country there is a River of Knowledge which has its fountain head high up in the Hills of Truth. This river is continually carrying down from the mountains the water from the Springs of Facts. To retain this water and swell the River of Knowledge, the Men of Industry and Study are building the great Dam of Science. Each group must keep its position up to the standard of the rest or be submerged and washed over the brink into the Abyss of Oblivion, while another group of builders or the adjoining ones fill in the place vacated.

The Central Station Builder is now reaching out and crowding the Isolated Plant Fellow, who will have to hustle to hold his place. The Specialist is building not so wide as some others, but higher, and is sufficiently ahead of the rest to be well out of danger. That the standard is being raised is everywhere evident. Science is constantly growing. The model power plant of a few years ago, with its then excellent equipment and commendable performance, is quite ordinary today, and there is as great a contrast in the engineers who operate them. *(Contributed by J. J. Ramsey, Wellington, Kan.)*

Power for Karpen Factory

By Thomas Wilson

SYNOPSIS—Review of installation built up from belt and rope drive to a central plant and electrical distribution. Choosing between a simple and compound engine. Steam is made for 19.6c. per 1000 lb., and in combination with the heating, current, costs 0.81c. per kw.-hr., including fixed and operating charges.

A maker of upholstered furniture, S. Karpen & Bros., of Chicago, has had an interesting experience in supplying power for the factory. Starting with line shafts in the various buildings, interconnected by rope drives and all turned by one large Corliss engine, this firm, after giving central-station current a year's trial, has finally installed an isolated plant and motors for individual and group driving. The plant was put in operation the first of the year and, as the records presented herewith will show, has been giving good results.

It was in May, 1899, that the company moved to its present location at the corner of Twenty-second and Union St. The site contained six acres of ground and at that time there were four three-story brick buildings of mill construction. In 1905, an annex to the mill and a new four-story building were built, bringing the total floor space up to 239,679 sq.ft. The Corliss was already pulling a friction load of 192 hp. out of a total peak load of 150 hp., and as the company did not wish to add to it, motors were placed in the new additions. With no generators in the plant, central-station current was given a tryout. In the annex the load averaged 150 amp. at 230 volts for 85 connected horsepower. In the new building 28 hp. in motors, 100 arc lights and 100 incandescent 16-cp. lamps gave an average demand for 275 amp. Independent meters for the incandescents, arcs and for the power load were put in. At that time, the incandescent lighting was billed at 20c. per kw.-hr., the arcs at 10 to 12c., depending on the demand, and the power load averaged about 4½c. Of course, these rates were exorbitant, and at the end of a year's time two 100-kw. generators were purchased. These units were belt-driven from the main shaft and up to last January have been supplying all electrical energy to the factory.

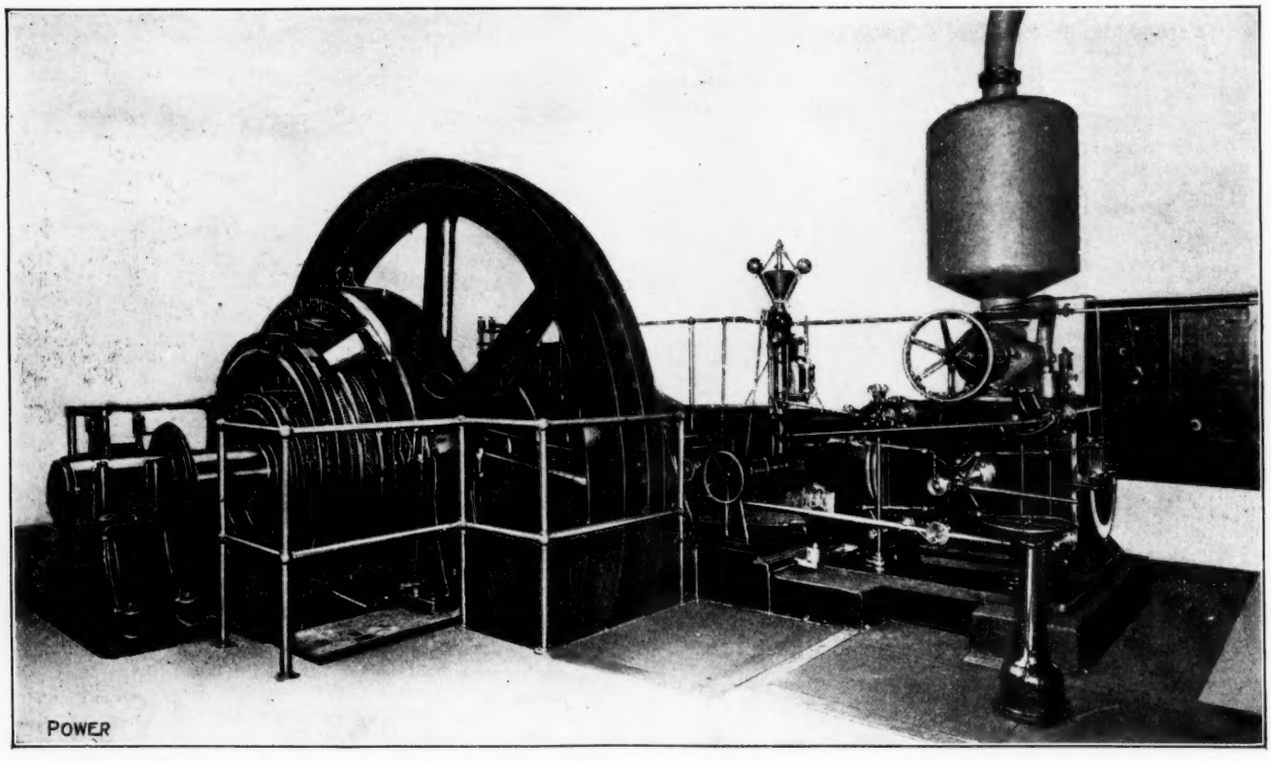


Fig. 1. Simple Corliss, 300-Kw. Unit

Boiler Plant

At the same time, a new boiler plant was installed, consisting of four 150-hp. return-tubular boilers, 72-in. by 18-ft., each containing 68 four-inch tubes. The boilers were built for 150-lb. pressure, but 125 lb. is the usual working pressure. The grates are of the shaking and dumping type, measuring 5½x5½ ft. The brick stack is 125 ft. high, 66 in. in diameter at the top and 72 in. at the bottom. Of the fuel used, 25 per cent. is shavings; these are chuted into the furnaces and burned with 6-in. shellbark egg coal. The latter is hand fired and a fuel bed of 6 in. is maintained. Over the fire the draft is 0.24 in. and in the breeching 0.78 in. These are the present conditions as the boiler plant was retained.

Use of Exhaust Steam

For heating the entire plant, there are 25,300 sq.ft. of direct radiation supplied by a two-pipe vacuum system. In the buildings there are eight small drying kilns fitted with 1700 sq.ft. of direct radiation. Outside are seven

larger kilns, in which air heated by 8000 sq.ft. of indirect surface is circulated by a fan driven by a 45-hp. motor. For cooking glue there are additional coils, aggregating 3844 sq.ft., which take live steam at 55 lb. pressure. The drying kilns and heating coils in the cabinet-making rooms utilize exhaust steam the year around, and in the four warmer months they take about 70 per cent. of the available supply. This was determined by weighing the condensation. During the balance of the year more or less heating is necessary throughout the entire factory, so that there is demand for all of the exhaust steam, and in the coldest weather some live steam in addition. During the day about 38 per cent. of the steam generated is used directly for manufacturing purposes.

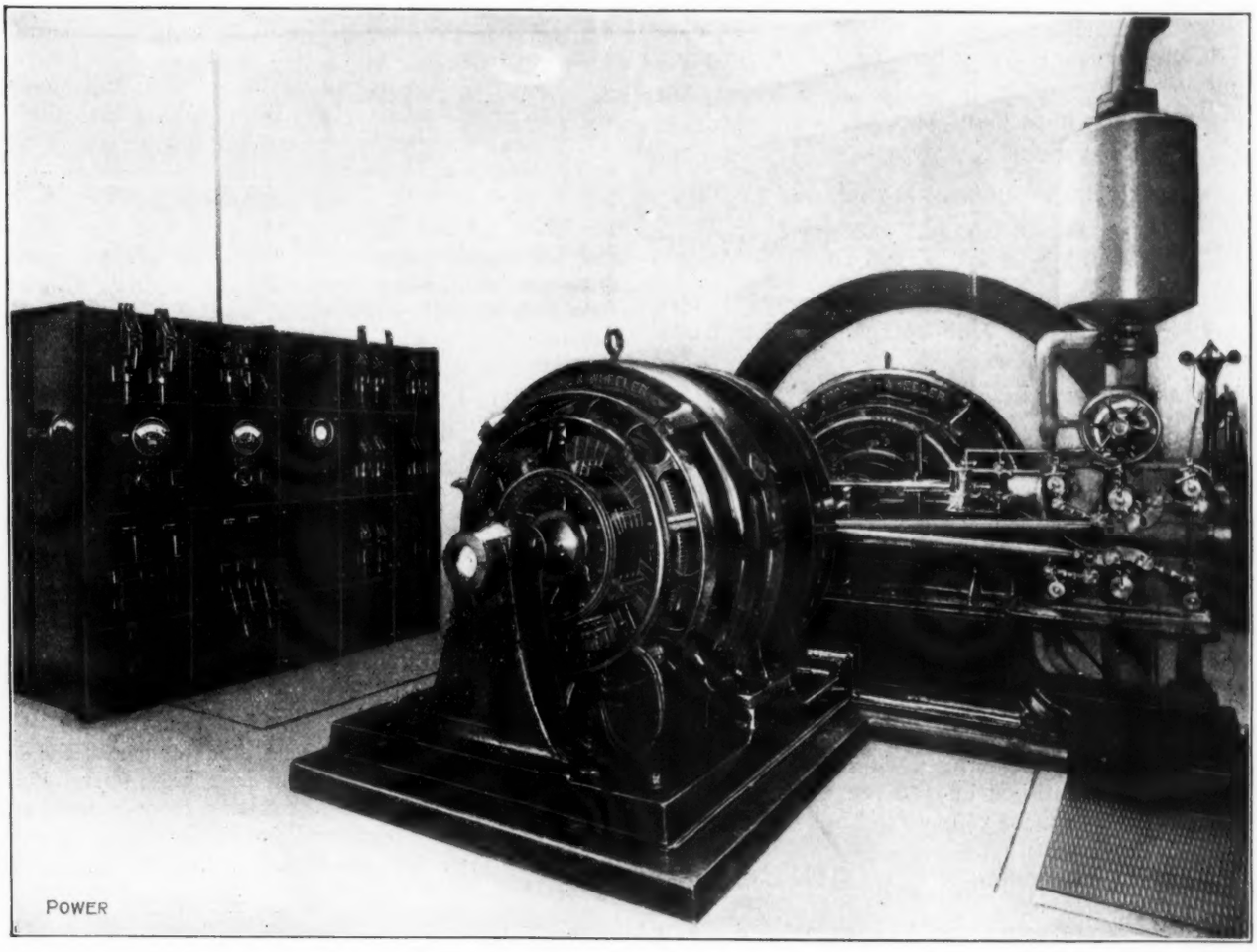


FIG. 2. FOUR-VALVE, 100-KW. UNIT AND SWITCHBOARD

INSTALL ISOLATED PLANT

As time wore on, the long lines of shafting and the rope drives began to wear out. A change was necessary, and the heavy friction loads of the past induced the company to put in electrical transmissions throughout. The central station again came forward, offering a minimum rate of 1.8c. and a maximum not to exceed 3c. Large demands for exhaust steam so influenced the cost of producing current, however, that a contract on this basis would have meant a large loss. A careful estimate, which has been verified by the records of the past six months, showed that the company could not afford to pay more than 1.3c. per kw.-hr. for its current. It was obvious that an isolated plant would soon pay for itself and in the latter part of 1913 the necessary generating equipment was purchased.

SIMPLE CORLISS OR COMPOUND FOUR-VALVE?

As soon as a plant had been decided upon, a discussion arose as to the proper type of prime mover. Frank Ory, the engineer, favored a simple Corliss making 100 r.p.m., arguing that although the slow-speed machine would be large for its capacity and costly, its mechanical efficiency would be high and the cost of upkeep a minimum. The engine would use more steam than a compound unit, but for the greater part of the time all of the exhaust steam would be utilized, so that the steam rate was of little consequence. A consulting engineer retained by the company favored a cross-compound four-valve engine, making 150 r.p.m., to obtain advantage of the lower steam rate in the summer months. Mr. Ory carefully analyzed the two cases and showed that his choice would save money in the course of a year, even with a higher steam rate and a cost for the unit $766 greater than that listed for the compound machine. His method of arriving at this conclusion may be of interest.

From his long experience in the plant, he determined that the average day load for the year would be 250 kw. and the peak load for three hours per day and four months per year during the heavy heating season would be 325 kw. He placed the evaporation at 7 lb. of water per pound

of coal and estimated that 25 per cent. of the water would be evaporated by shavings and floor sweepings. Thus 75 per cent. of the fuel would be coal costing $3 per ton in the bunker. A steam rate of 24 lb. per i.hp.-hr. was quoted for the simple engine and 20.5 for the compound. For the simple Corliss, efficiencies of 93 per cent. for the engine and 92 per cent. for the generator were taken, and for the compound 88 and 90 per cent., respectively. The lower mechanical efficiencies for the compound unit were due to cheaper construction, two cylinders instead of one and a lighter generator provided with less copper.

Assuming that only coal is burned under the boilers for 10 hr. per day and 300 days per year, the fuel cost for the steam going through the simple engine to supply the heating system besides generating 250 kw. for power, would be

$$\frac{250 \times 10 \times 300 \times 24 \times \$3}{7 \times 2000 \times 0.93 \times 0.92 \times 0.746} = \$6040$$

For the compound engine the cost would be

$$\frac{250 \times 10 \times 300 \times 20.5 \times \$3}{7 \times 2000 \times 0.88 \times 0.90 \times 0.746} = \$5577$$

The peak load is 75 kw. more than the average for three hours per day and four months per year, so that the following amounts should be added:

For the simple engine:

$$\frac{75 \times 4 \times 3 \times \$6040}{250 \times 12 \times 10} = \$181 \qquad (A)$$

and for the compound engine:

$$\frac{75 \times 4 \times 3 \times \$5577}{250 \times 12 \times 10} = \$167 \qquad (A)$$

The $6040 and the $5577 are the yearly totals for 250 kw. and exhaust steam for heating. For eight months the charge against the simple engine would be:

$$\$6040 \times \tfrac{8}{12} = \$4027 \qquad (B)$$

and for the compound engine

$$\$5577 \times \tfrac{8}{12} = \$3718 \qquad (B)$$

For the four warm months, when 70 per cent. of the exhaust steam is utilized and 30 per cent. lost to atmosphere, the charge for 250 kw. and heating would be:

For the simple engine,

$$\$6040 \times 0.70 \times \tfrac{4}{12} = \$1409 \qquad (C)$$

For the compound engine,

$$\$5577 \times 0.70 \times \tfrac{4}{12} = \$1301 \qquad (C)$$

The cost of the 30 per cent. exhaust steam wasted would be:

For the simple engine,

$$\$6040 \times 0.30 \times \tfrac{4}{12} = \$604 \qquad (D)$$

For the compound engine

$$\$5577 \times 0.30 \times \tfrac{4}{12} = \$558 \qquad (D)$$

It is evident that the cost for the year equals the sum of A, B, C and D. Omitting D would give the cost when all of the steam is used for heating. For the simple engine this would be

$$\$181 + \$4027 + \$1409 = \$5617$$

and for the compound engine

$$\$167 + \$3718 + \$1301 = \$5186$$

At the working pressure, 140 lb. abs., the heat contents in a pound of steam is 1192.2 B.t.u. In expanding in the engine to a back pressure of 16 lb. abs., 158.1 B.t.u. are transformed into work, the balance, 1034.1 B.t.u., is available for heating. Then the actual cost of coal for power in the simple engine is

$$\frac{158.1}{1192.2} \times \$5617 = \$745 \qquad (E)$$

and in the compound engine

$$\frac{158.1}{1192.2} \times \$5186 = \$688 \qquad (E)$$

To these figures should be added the cost of coal for the 30 per cent. of the steam rejected during four months of the year. This would make the total coal cost for power

$$\$604 + \$745 = \$1349$$

for the simple engine and

$$\$558 + \$688 = \$1246$$

for the compound engine. These figures are based upon using coal only for fuel, but 25 per cent. is shavings, so that the cost for coal would be 75 per cent. of the above amounts, or $1011.75 and $944.50, respectively.

For fixed charges, 5 per cent. for interest and 2 per cent. for insurance and taxes were allowed in each case. Due to higher speed, less weight and more bearings, depreciation for the compound was placed at 7 per cent. For the simple engine 5 per cent. was allowed. The cross-compound, being practically two engines, will use more oil, at least $100 worth, and this was charged against it.

Table 1 is a summary of the fixed charges, fuel cost and additional oil, labor and supplies being considered

TABLE 1. SUMMARY OF FIXED CHARGES AND FUEL COSTS

	Simple Corliss		Compound Four Valve
Cost of Unit.....	$11,966.00		$11,200.00
Interest, 5%.....	$598.30	at 5%	$560.00
Ins. and taxes, 2%...	239.32	at 2%	224.00
Depreciation, 5%....	598.30	at 7%	784.00
Fixed charges	$1,435.92		$1,568.00
Cost of fuel.....	1,011.75		944.50
Cost extra oil.....			100.00
Total	$2,447.67		$2,612.50

the same in each case, although the cost of supplies for the compound engine should be slightly greater.

It is evident that the difference in favor of the simple Corliss is $164.83, and it would have been greater if a higher-grade compound engine costing more money had been selected. The compound would also have required the use of more live steam for heating in the colder

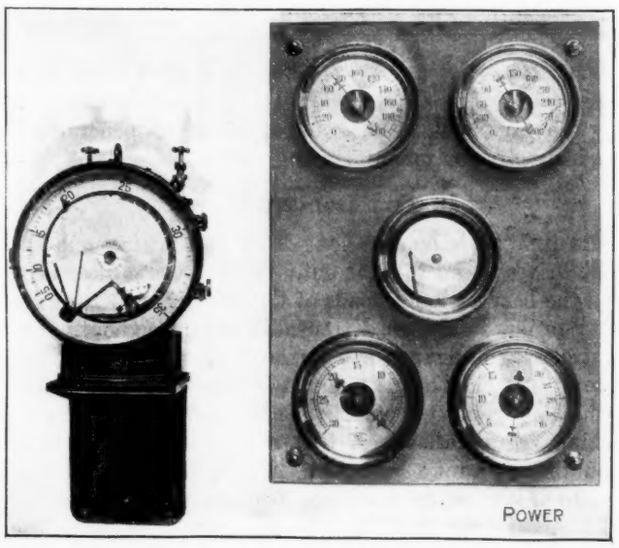


FIG. 3. FLOW METER AND GAGE BOARD

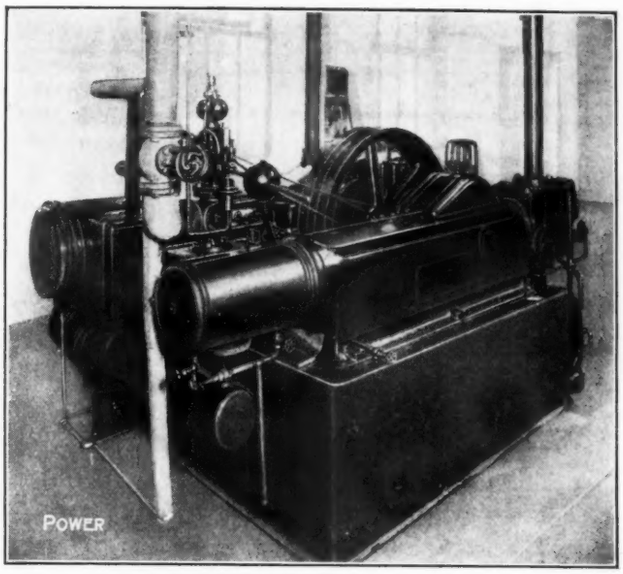


FIG. 4. COMPOUND STEAM TWO-STAGE AIR COMPRESSOR

weather. Naturally, a simple Corliss engine was set up and put into operation the first of the year.

ENGINE ROOM

The main unit to carry the day load consists of a simple full-stroke releasing Corliss engine, 24x36-in. and 100 r.p.m., directly connected to a 300-kw. two-wire generator delivering current at 230 volts, Fig. 1. A 15x17-in. four-valve engine driving a 100-kw. generator at 200 r.p.m handles the night and Sunday loads, Fig. 2. The day load averages about 1100 amp., which at 230 volts would make 253 kw., but the peak load runs as high as 1800 amp., or 414 kw. The large machine easily cares for the overload. At night the average load is not more than 150 amp., so that the small unit is underloaded. Through the day there is plenty of exhaust steam until the temperature gets down to 10 deg. above zero, but at night, with the light load and the heavy demands by the kilns, live steam is required the year around.

A modern switchboard, a water-flow meter, a gage panel, Fig. 3, for live, exhaust steam and air pressures are part of the engine-room equipment. There is also a compound-steam two-stage air compressor supplying air at 80 lb. pressure for factory use. Its capacity is 325 cu. ft. of free air per min. at a speed of 225 r.p.m.

The large engine has a continuous oiling system consisting of a pump driven by the rocker arm of the engine and a small combination separator and filter mounted in a concrete container and attached to the engine base below the floor. As the oil returns from the various bearings it enters the separator, passes through the filter and is pumped to a storage tank mounted on the wall of the engine room. From here it flows by gravity to points of supply. Force-feed pumps lubricate the cylinders of both engines.

SMALL LIVE-STEAM PIPING

A feature of interest is the small steam piping to the engines and large receiver-type separators having volumes four times greater than the engine cylinders. The drop in pressure between boiler and cylinder at full load is only 4 lb. For the large engine a 5-in. pipe rises from a 12-in. header a distance of 3 ft., runs horizontally 20 ft. and drops 5 ft. into the top of the receiver. The steam passes through an 8-in. throttle valve into the cylinder. The exhaust pipe is 9 in. in diameter to the first elbow, then 10 in. to a 14-in. main supplying the heating system or discharging to atmosphere through a 12-in. riser. For the small unit the supply pipe is 3 in. in diameter, the throttle valve 5 in. and the exhaust pipe 6 in. The supply pipe to the large engine has 0.0667 sq.in. of area per kilowatt of rating and the exhaust pipe 0.2093 sq.in. For the 100-kw. unit the pipe areas per kilowatt of rating are 0.0739 and 0.2889 sq.in., respectively.

The large engine is rated at 500 hp. and the small one at 150 hp., the steam rates at full load being 24 and 28 lb. per i.hp.-hr., respectively, or 26.6 and 31.1 per b.hp.-hr. The former would then require at full load

$$\frac{500 \times 26.6}{60} = 221.7 \ lb.$$

of steam per minute, and the small engine

$$\frac{150 \times 31.1}{60} = 77.7 \ lb. \ per \ min.$$

At an average pressure of 123 lb. gage, the volume of each pound of steam would be 3.27 cu.ft. The volumes of steam per minute passing in each case would be

$$221.7 \times 3.27 = 725 \ cu. \ ft.$$
$$77.7 \times 3.27 = 254 \ cu. \ ft.$$

Assuming continuous flow, as is usual, the velocity in the supply pipe of the 500-hp. engine would be 5220 ft. per min., and at atmospheric pressure the exhaust velocity would be about 11,600 ft. per min. For the small engine the velocities would be 4950 ft. per min. for the supply and 8820 ft. per min. for the exhaust. The small drop in pressure shows that the pipes are ample, and the good results may be attributed to the large separators.

OPERATING RECORDS

Table 2 gives the costs of operating the plant for the first six months as computed by the engineer. It will be noticed that steam, whether live or exhaust, for pur-

TABLE 2. COST OF OPERATING POWER PLANT FROM JAN. 1 TO JULY 1, 1914

BOILER ROOM		
Coal, 4,022,363 lb., at $2.60 per short ton	$5,229.07	
Labor, 3 firemen	1,000.00	
Ash removal	130.20	
Heating boiler water, 160° to 212°	853.84	
Water, washing out boilers, etc.	40.00	
Lighting boiler room	25.00	
Blowing down boilers	50.64	
Fixed charges on $6000, int. 5%, dep. 5%, ins. ½ of 1%	315.00	
Total boiler room costs		$7,643.75
Water evaporated, lb.	40,614,458	
Cost evap. 1000 lb. of water, c.	19.6	
Water evap. per lb. of coal, lb.	10.09*	
ENGINE ROOM		
Labor, 2 engineers and one-half chief's time	$1,850.08	
Oils	112.64	
Supplies	248.39	
Water for eng. room, toilet and shower	35.00	
Light for engine room	25.93	
Steam rejected to atmosphere	178.44	
Fixed charges, 10½% on $17,000	892.50	
Total engine room costs		$3,342.98
Engine and boiler room costs		$10,986.73
CREDITS		
Steam for fire pump at 20c. per 1000 lb.	$199.12	
Steam for heating and manufacturing at 20c. per 1000 lb.	7,088.56	
Steam for compressed air at 20c. per 1000 lb.	573.44	
Total		$7,861.12
Cost of generating electricity		$3,125.61
Kw.-hr. generated		386,100
Cost per kw.-hr., c.		0.81

*Includes water evaporated by shavings, actual evaporation by coal $0.75 \times 10.09 = 7.57$.

PRINCIPAL EQUIPMENT OF S. KARPEN & BROS. FACTORY PLANT

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
4	Boilers	Return tubular	72-in. x 18 ft.	Generate steam	Natural draft, 125 lb. gage	Frost Mfg. Co.
4	Grates	Dumping and shaking	5½x5½ ft.	Boiler furnace		Economic Engineering Co.
2	Pumps	Simplex	10x6x12-in. and 12x7½x12 in.	Boiler feed water	125 lb. steam pressure	American Steam Pump Co.
2	Pumps	Simplex	6x8x12-in. and 8x10x12-in.	Vacuum heating system	125 lb. steam pressure	American Steam Pump Co.
1	Heater	Open	1000 hp.	Heat feed water	Exhaust steam	Warren Webster & Co.
1	Fire pump	Duplex	18x10x12-in.	Fire protection	1000 gal. per min. at 70 strokes	Fairbanks Morse & Co.
1	Meter	Water flow		Recording boiler feed		General Electric Co.
1	Engine	Simple Corliss	24x36-in.	Main unit	125 lb. steam, atm. exh., 100 r.p.m.	Nordberg Mfg. Co.
1	Generator	Direct current	300 kw.	Direct connected to Corliss	230 volt, 100 r.p.m.	Crocker-Wheeler Co.
1	Engine	Four valve	15x17	Main unit	125 lb. steam, atm. exh., 200 r.p.m.	Chuse Engine & Mfg. Co.
1	Generator	Direct-current	100 kw.	Direct connected to four-valve	230 volt, 200 r.p.m.	Crocker-Wheeler Co.
1	Air compressor	Compound steam. Two-stage air	8x13x12-in. 7½x12x12-in.	Air at 80 lb. to factory	125 lb. steam, 225 r.p.m.	Ingersoll Rand Co.
4	Gages	Indicating	10-in. diam.	Steam and vacuum		Bogardus Co.
1	Gage	Recording	10-in. diam.	Boiler pressure		Bogardus Co.
2	Ammeters		One 2000 and one 600 amp.	Generator current		Weston Electrical Instrument Co.
1	Voltmeter	Swinging bracket	600-volt	Generator voltage		Weston Electrical Instrument Co.
1	Watt-hour meter	Recording		Record output		Sangamo Electric Co.

poses other than generating electricity are properly credited at the cost of production. Much of the steam credited to heating has already passed through the engines, but without the engines live steam would be needed to take its place, so the credit is legitimate. The same plan is followed out in charging the exhaust steam used for heating the feed water and that wasted to atmosphere. This procedure is unusual, but consistent with the heating charge. The item for blowing down boilers would seem to be already covered by previous charges, but it is so small that results will not be influenced much either way.

A cost per 1000 lb. of steam of 19.6c. is fair for a hand-fired plant, and a cost of 0.81c. per kw.-hr., including both operating and fixed charges, is excellent—in fact, much less than the central stations' cost of production. If current had been purchased, the services of one engineer only, amounting to $480 for the six months, could have been dispensed with. The fixed charge item on the generating equipment, most of the oil and supply items and the rejected steam charge would also be eliminated. This makes a total of

$$\$480 + \$892.50 + \$178.44 + \$248.39 + \$112.64 = \$1911.97$$

Dividing by 386,100 kw.-hr., gives 0.49c. Adding this to the 0.81c. makes a total of 1.3c. per kw.-hr., the maximum the company could afford to pay for central-station service. When this is compared to a minimum rate of 1.8c. increased to approximately 3c. by the numerous charges usually annexed, it is plain that an isolated plant in this case is paying big dividends.

Boring Engine Cylinders

The accompanying photographs illustrate a boring bar designed for convenient use in the power plant. It may be used for boring engine cylinders, Corliss engine valve ports, crosshead slides, bearings that have been rebabbitted, etc. Fig. 1 shows the bar complete except for the crank-end bearing frame which, although furnished with each bar, is not ordinarily employed when boring an engine cylinder, because this bearing is fixed in the piston-rod stuffing-box. The cutter head *A* is shaped like a four-point star. Three of the arms carry the regular cutting tools while the fourth is designed to hold a tool at 45 deg. so that the point is in advance of the face of the cutterhead when making the cylinder counter bore.

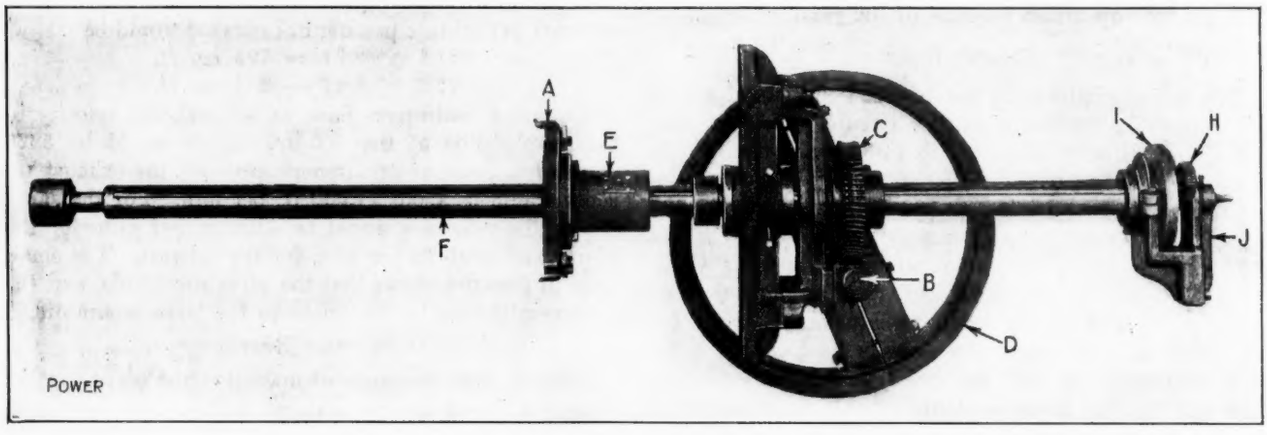


FIG. 1. BORING BAR, SHOWING DRIVING MECHANISM

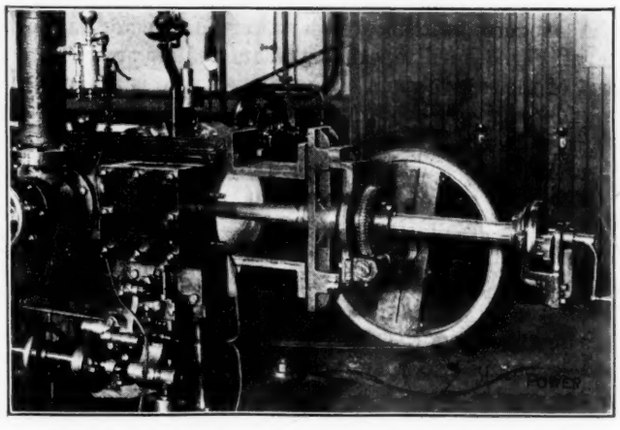


FIG. 2. BAR IN POSITION IN ENGINE CYLINDER

The bar is revolved by the worm *B* and wheel *C* driven from any power source through the pulley *D*. The worm and wheel are used to produce a steady motion, free from chattering, and to insure a smooth clean cut. The cutter-head carrier *E* is fitted with two lugs or keys which slide in races cut in the side of the bar. Thus, while the carrier may slide freely along the bar, it must revolve with it.

The carrier is fed backward or forward by means of the long screw *F* turning in a slot in the side of the bar and meshing with a half-nut fixed on the inside or bearing surface of the carrier. The feed screw is made to revolve automatically by means of a train of gears, the ratios of which can be changed to produce any rate of feed from $^1/_{64}$ to $^1/_4$ in. per revolution of the boring bar.

At the right of the photograph the large gear *H* is fixed to the disk *I* which is integral with the bar. Thus the gear moves in a circle when the bar revolves. It meshes with a gear secured to the frame *J* which hangs on a loose collar on the end of the bar and, hence, does not revolve with the bar. Another gear on the same pin as gear *H* drives a gear fixed on the end of the feed screw.

For protection and adjustment at each bearing, the bar is fitted with a movable sleeve which takes all the wear. This sleeve and the bearing are slightly tapered so that the wear can be taken up by shifting the sleeve.

The bar is made by Boxill & Bruel, 848 West Sixth St., Cincinnati, Ohio, in two standard sizes for cylinders from 4 to 16 in. and from 16 in. to 60 in. The standard lengths are 6 and 12 ft., but special sizes can be furnished. When a plant is not large enough to warrant the purchase of a bar, the manufacturer will lease one for any desired length of time.

The Effect of Compression

By Robert C. H. Heck

SYNOPSIS—A digest of several sets of experiments upon the effect of compression on steam-engine economy, with conclusions that a small amount of compression is desirable, but should not increase with earlier cutoff; also, that wasteful effect of excessive compression is greater with light load than with full load. The tests seem unanimous in recommending about two units of compression for noncondensing engines, and rather more for condensing engines. All the data referred to indicate that clearance steam is by no means free from moisture at the beginning of compression.

Following the pioneer work of Dwelshauvers-Dery and the Duchesnes at Liège, several sets of careful and extensive experiments upon the effect of compression on the economy of the steam engine have been made in different European college laboratories. The recent publication of the last of these, in January of the current year, makes this an appropriate time for presenting a digest of them all. Introductory particulars of the tests to be reviewed are as follows:

A. By Prof. Dwelshauvers-Dery, at the University of Liège, published in *Revue de Mécanique*, 1897, Vol. 1, page 939; see also Power, June 28, 1910, page 1157. Four-valve engine, cylinder 300x600 mm. (11.8x23.6 in.), clearance 0.066. Engine run noncondensing, with steam of about 80 lb., absolute pressure, superheated 80 deg. F.; steam jackets not in action; speed varied.

B. By Dr. H. Klemperer, at the Dresden Technical High School, in 1903; published in *Zeitschrift des Vereines deutscher Ingenieure*, May 20, 1905, pages 797-807. Corliss engine, steam-jacketed on barrel and heads, cylinder 180x450 mm. (7.1x17.7 in.), extended piston rod 1.57 in. diameter on both sides of piston, clearance 0.045, speed 102 r.p.m. Engine run condensing with steam of 100 lb. absolute pressure, noncondensing with 114 lb. absolute.

C. By Prof. J. Boulvin, at the University of Ghent, published in *Revue de Mécanique*, 1907, Vol. 20, pages 109-136. Corliss engine, valves in cylinder heads, full steam jackets; cylinder 250x500 mm. (9.8x19.7 in.); clearance 0.038, speed 102 r.p.m.; run noncondensing with saturated steam of about 92 lb. absolute pressure.

D. By E. Heinrich, at the Stuttgart Technical High School, published in *Zeit. Ver. deutsch. Ing.*, Jan. 3, 1914, pages 15-22 and Jan. 10, 1914, pages 58-62. Lift-valve engine, full steam jackets, cylinder 250x760 mm. (9.8x30 in.), clearance 0.087; r.p.m., 92.5; engine run condensing with saturated steam of 116 lb. absolute, and with an absolute pressure of about 2.2 lb. below the exhaust valves.

Each experimenter has gone into a more or less elaborate thermodynamic analysis of his observations and results. The present purpose, however, is to consider no more than the principal, primary determination of steam consumed per indicated horsepower-hour. But before giving and comparing the results of experiment, a simple rational deduction will be worked out, based on three good average empirical assumptions; and it will be found that the conclusions derived from this preliminary discussion will agree very well in form with the experimental curves. The three assumptions to be made are:

(a) That equilateral hyperbolas well enough represent the average form of actual expansion and compression curves.

(b) That the clearance steam may be taken as dry saturated at the beginning of compression.

(c) That for a given cylinder, engine speed, and pair of limiting pressures, the missing steam at cutoff will be constant in amount, or independent of cutoff and compression.

From each of these, actual performance may depart pretty widely; but in general, the departures are quantities of the second order of magnitude, whose study belongs to a close and detailed analysis.

Theoretical Deduction

In Fig. 1, outline *ABCDEFA* is a steam or indicator diagram which is ideal in that it shows none of the losses of pressure and the rounding of corners due to pipe, port and valve action. But the steam is supposed to be under the full thermal influence of the cylinder wall, so that

the amount present at cutoff and actually filling the space *PB* would have the volume *PG* if none of it were condensed. Under assumption *(a)*, curves *BC* and *EF* are of the form $pv = C$.

Now, the whole weight of steam substance present at cutoff and during expansion is made up of two parts, the clearance steam caught and compressed from the preceding cycle and the new steam supplied from the boiler for the present cycle. If the former is known, its full, dry volume can be laid off as *PH*, and its volume as a portion of the actual steam-and-water mixture at cutoff will be *PJ*. In this particular diagram, *PB* is 0.7 of *PG*, hence also *PJ* is 0.7 of *PH*. The initially condensed

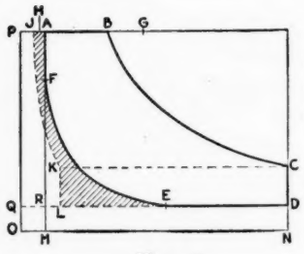


FIG. 1

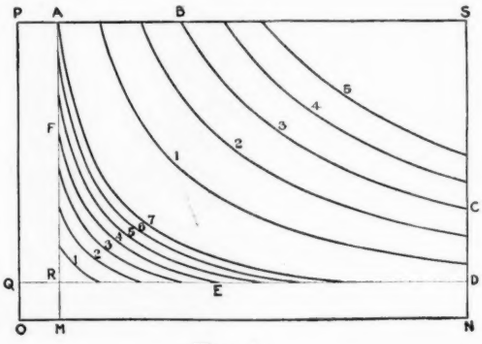


FIG. 2

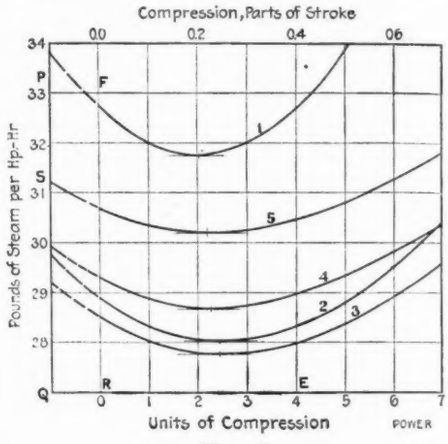


FIG. 3

Fig. 1. Ideal steam diagram, with separate steam expansion curve for the clearance steam. Fig. 2. Group of ideal diagrams, with varying cutoff and compression. Fig. 3. Curves showing variation of steam consumption with compression.

or "missing" steam, which would be represented by volume *GB*, is then 30 per cent. of the total steam. As cutoff and compression vary (see Fig. 2), this quantity *BG* is to remain constant, not relatively or as a percentage, but absolutely.

If the whole steam measured by volume *PG*, Fig. 1, were admitted to a cylinder without any clearance space at all, while yet suffering the shrinkage *GB* because of wall action, it would perform the work represented by area *PBCDQP*. By drawing an equilateral hyperbola *JK*, like *BC* and ending at the same terminal pressure, the whole work area is divided into proportional parts, *PJKLQP* for the weight of clearance steam and *JBCDLKJ* for the weight of fresh steam.

In the actual engine, with piston displacement beginning at *MA* instead of *OP*, the net work to be credited to the supplied steam is area *ABCDEFA*. Then the shaded area between *EFA* and *JKL* represents the loss of possible work output due to clearance and compression.

The measurement of steam consumed in a given time, from which can easily be computed the steam supplied to the cylinder per cycle, is one of the main determinations in an engine test. The weight of clearance steam cannot be measured, but must be inferred from the indicator diagram, by methods which at best have in them some degree of uncertainty. Assuming dry saturation at *E*, its weight is found from volume *QE* and pressure *OQ*.

Whatever the actual behavior of the clearance steam during compression, and whether or not curves *BC* and *EF* are really of the same form, the average quality or degree of dryness is higher during compression than during expansion. Therefore, when a separate expansion curve *JK* is drawn for the weight of clearance steam, it will lie within or to the left of curve *FE*. The earlier assumption that a curve like *JK* will agree with *FE*, or that in expansion the clearance steam will give back all the work expended in compressing it, is erroneous and leads to false conclusions.

Group of Ideal Diagrams

In Fig. 2 is shown a combination of 40 diagrams like Fig. 1. The five expansion curves are marked by numbers which indicate cutoff in tenths of the stroke. The numbers on the compression curves similarly show compression *ER* in tenths of stroke *NM*. Including zero, there are eight compressions to be combined with each of the five cutoffs.

Now any area *ABCDEFA* represents work done. From a volume got by adding a constant *BG* (Fig. 1) to the variable *PB* can be computed the weight of total steam, and from volume *QE* the weight of clearance steam. It is then not difficult to calculate steam consumption *S* in pounds per horsepower-hour, and most simply by letting stroke *MN* represent a piston displacement of 1 cu.ft. The results of this calculation are laid out in Fig. 3, where each curve is for a particular cutoff and a horizontal tangent determines the best compression (giving lowest steam consumption) with each cutoff.

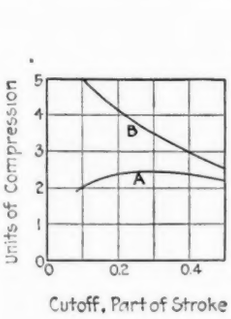


FIG. 4. CURVES OF BEST COMPRESSION

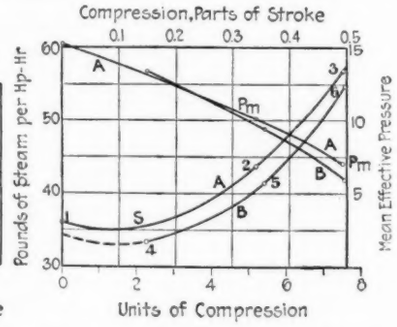


FIG. 5. DWELSHAUVERS-DERY STEAM-RATE RESULTS

Variation of Steam Consumption with Compression

The curves, Fig. 3, really begin at end line *RF* (Fig. 2) for zero compression; but the dotted extensions show what would be the effect of a complete emptying of the clearance space; if this were possible, it would make *PG*

(Fig. 1) the required amount of fresh steam. The unit of the scale at the bottom of Fig. 3 is the clearance volume. Because clearance was taken as 10 per cent. in Fig. 2, "units of compression" here agree with tenths of stroke; but generally the two scales are different, as appears in Figs. 5, 7, etc.

Curves of Best Compression

A very important showing of Fig. 3 is that the best compression is not large, ranging from 2 to 2.5 units for the determining conditions of these diagrams, and that it varies but little with the amount of expansion. The plot of best compressions in Fig. 4, on cutoff as base, brings this out even more clearly. Curve *A* is from Fig. 3, while curve *B* is got from Fig. 2 and a group of curves like Fig. 2, but based on the assumption that expansion *JK* in Fig. 1 will coincide with compression curve *EF*. Subject to secondary influences which will modify it in detail without changing its essential form, curve *A* (Fig. 4) represents actual engine working and shows that, instead of continually increasing as cutoff becomes earlier, the most advantageous compression is nearly constant over a wide range and grows less for very early cutoff, and from Fig. 3 it appears that the harm done by wrong compression is relatively greater with early than with late cutoff.

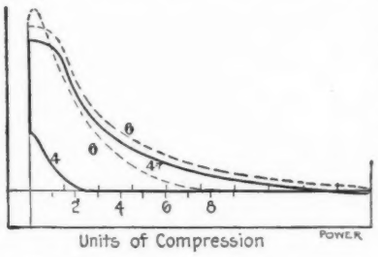


Fig. 6. Indicator Diagrams, Tests Nos. 4 and 6 of Fig. 5.

Review of Experiment

All the experiments to be presented were made in the metric system, but have been reduced to English units.

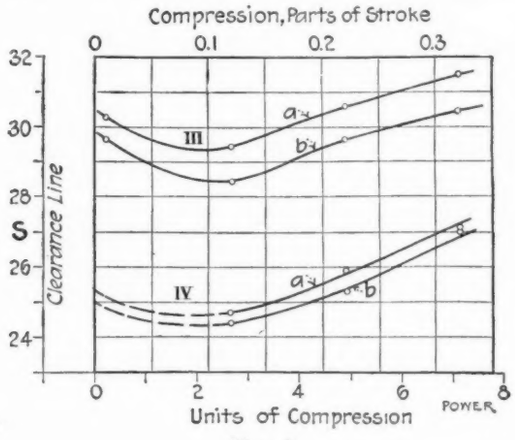


Fig. 7

Klemperer's steam rates: Noncondensing, no steam in jackets, 114 lb. absolute steam pressure; Series III, saturated steam; Series IV, steam superheated about 90 deg. F.; groups a, cutoff 0.20, groups b, cutoff 0.30.

The two important relations between units are that 1 kg. per sq.cm. = 14.22 lb. per sq.in., and that 1 kg. per metric hp. = 2.235 lb. per English hp.

Dwelshauvers-Dery Tests

The six tests reported by Prof. Dwelshauvers-Dery are represented by Figs. 5 and 6. The principal curves in Fig. 5 show steam rate *S*, while the m.e.p. or p_m is also given, chiefly to emphasize the lightness of the load on the engine. Cutoff was made so early (at 0.1 of the stroke) because full expansion to exhaust pressure was desired; the engine would have a better steam rate with fuller admission. The tests are in two groups, Nos. 1 to 3 made at 45 r.p.m., and Nos. 4 to 6 at 62 r.p.m.

In laying out the curves of Fig. 5, the writer has not followed those of the experimenters in their published report. They give the amounts of compression as zero for No. 1, 0.1 for No. 4, 0.3 for Nos. 2 and 5, and 0.4 for Nos. 3 and 6. But the indicator diagrams do not show these nominal values to have been realized; in Fig. 6, for instance, the actual compression is about 0.15 for No. 4 and nearly 0.5 for No. 6. The original curves of steam rate *S* dropped continually from high compression toward zero compression, seeming to show that any least degree of compression was harmful. The curves here drawn, with a minimum of steam rate at about 1.5 units of compression, better represent the experiments and are in accord with the other results, and with reason.

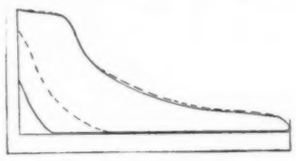


Fig. 8

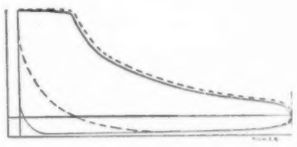


Fig. 10.

Fig. 8. Indicator diagrams line IIIa in Fig. 7; cutoff, 0.20; compression, 0.12 and 0.32. Fig. 10. Indicator diagrams, line Ia in Fig. 9; cutoff, 0.23; compression, 0.085 and 0.50.

Klemperer Steam Rates: Noncondensing

The principal groups of Klemperer tests are given in Figs. 7 and 9, with atmospheric exhaust and with vacuum, respectively; there is a further differentiation between saturated and superheated steam in Fig 7 and between the use and nonuse of steam jackets in Fig. 9.

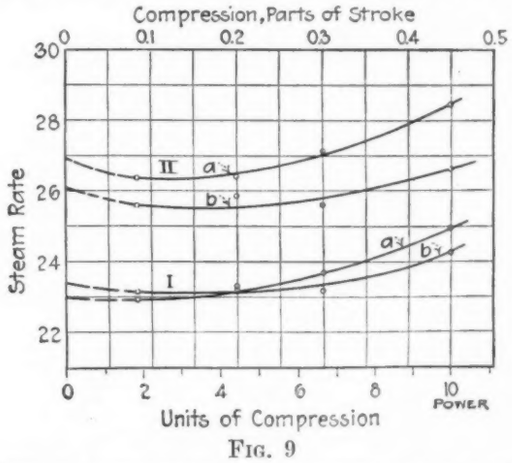


Fig. 9

Klemperer condensing tests: Saturated steam at 100 lb. absolute pressure; Series I, steam in jackets; Series II, no steam in jackets; exhaust pressure, 2.6 lb. absolute; groups a cutoff 0.12; groups b, cutoff 0.23.

With steam jackets in action, the steam used in them is included in the steam rate *S*. The earlier cutoff is short of that for best economy in both sets of tests, as is shown by curve *a* lying from 2 to 3 lb. above curve *b* at III and II; but the tendency of both economy-promoting expedients—that is, of superheating and of steam jacketing—to neutralize the bad effects of too early cutoff is shown by the approach of curve *a* to curve *b* at IV and I.

The curves of Fig. 7 are like those of Fig. 3 in having their minima at about two units of compression. As is reasonable, the best compression when running condensing (Fig. 9) is greater, ranging from 3 to 4 units. Dr. Klemperer ran some other tests which it is not worth while to reproduce here. In one case, the cutoff was made so early as to give complete expansion, and the curve of steam rate is of the same form as those drawn in Fig. 6. In another series, the clearance volume was increased, by setting out the cylinder heads, from 4.5 to 14.5 per cent.: this considerably increased the best amount of compression, when measured in parts of the stroke, but made it less in terms of multiples of the clearance volume.

Boulvin Tests with Steam Jackets

The tests by Prof. Boulvin, Figs. 11 and 12, are in groups with compression kept constant and cutoff varied; then the primary curves, Fig. 11, are each for a particular value of compression, expressed and marked in per cent. of stroke. Having these, it is a simple matter to take intersections of the curves with a line of uniform cutoff and project them over, as from *A* to *B*, upon ordinates located by degrees of compression. This gives the curves in Fig. 12, similar in terms to Figs. 3, 5, 7 and 9, and designated by cutoff values in per cent of stroke

Two anomalies appear in this group of tests. The first is the inconsistent shape of the curve for 3 per cent. of compression in Fig. 11: from the way that it cuts across the other curves, at such a decidedly different slope, results the peculiar shape of the curves in Fig. 12 at their left-hand ends. However, since there are but three experimental points to determine primary curve 3, it may be considered an erratic variation. Another peculiar point is the marked reversal of curvature to the right of the minima in Fig. 12. A trace of the same tendency appears in curves III of Fig. 7, but further experimental evidence would be needed to establish such a shape of curve as typical or characteristic.

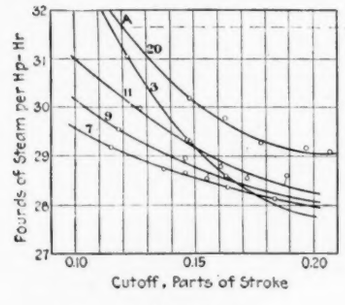


Fig. 11

Boulvin tests, plotted as made with compression held constant and cutoff varied.

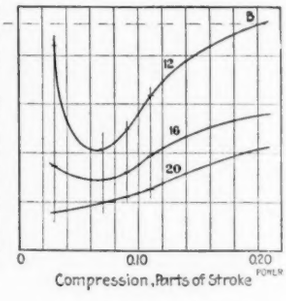


Fig. 12

Curves derived from Fig. 11 with compression varying under constant cutoff.

Steam pressure, 92 lb. absolute; r.p.m., 102; clearance, 0.038; steam jackets in action and jacket steam included in steam consumption; exhaust at pressure of atmosphere.

Boulvin Tests without Steam Jacketing

A second group of Boulvin tests, made without steam in the jackets, is represented by Figs. 13 and 14, which are similar in scheme and relation to Figs. 11 and 12. Having only three determining points, there is much freedom as to the shape of the derived curves in Fig. 14, and they are made consistent with Fig. 3 in shape, rather than with Fig. 12. Clearance being 3.8 per cent., the minima of steam consumption lie at about two units of compression in both Fig. 11 and Fig. 13.

Heinrich Tests

The Heinrich tests, directly diagrammed in Fig. 15, were run with different compressions under constant cutoff. A special variable was the amount of clearance surface, which was changed by fastening to the cylinder heads perforated cast-iron plates, separated by thin distance blocks. The engine has the very unusual feature of a crank of variable throw, such as is commonly used in certain machine tools. In these experiments, the stroke was shortened from the full 30 in. to as little as 27.6 in., being adjusted so as to keep the clearance-volume ratio the same for each amount of artificial surface. As noted beneath the diagram, the open-circle points and full-line curves show tests made with steam jackets, the upper group of points and curves tests without jackets. Within these groups, the varying condition was amount of surface, which was about as follows:

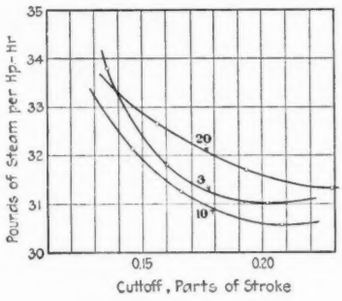


Fig. 13

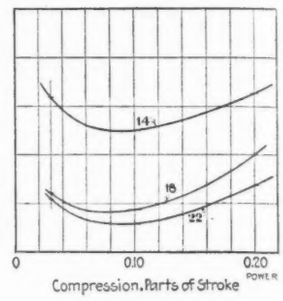


Fig. 14

Fig. 13. Boulvin tests without steam jacketing, other conditions as in Fig. 11. Fig. 14. Curves derived from Fig. 13 with compression varying under constant cutoff.

Curves	Clearance Surface
1 and 6	4.82 sq.ft.
2 and 7	6.11 sq.ft.
3 and 8	7.61 sq.ft.
5, 5 and 9	8.96 sq.ft.

Curves 4 and 5 are for the same arrangement of surface, but differ because the exhaust pressure was 2.12 lb. for No. 4 and 2.79 lb. for No. 5.

The main points of interest in the present connection may be summed up in two brief statements concerning Fig. 15, namely:

Along with the smaller steam rate, when the jackets are in action, goes the fact that the curves are flatter, showing that the harmful influence of excessive compression is less as the conditions within the cylinder are better.

The influence of compression is small in this condensing engine, but there is a well established minimum of steam rate at about 2 units of compression without jackets and at nearly 3 units with jackets.

Conclusions

The experimental curves of steam consumption, with compression as base, are generally similar to those laid out in Fig. 3. They confirm the deduction that a small amount of compression is desirable for steam economy: but this should not increase with earlier cutoff, and the wasteful effect of excessive compression is greater with light load than with full load on the engine.

For noncondensing engines, the experiments are remarkably unanimous in recommending about two units of compression, or enough to raise the terminal pressure

(at end of compression) to 30 or 40 lb. above atmosphere. For condensing engines, the best length of compression is greater, but the resulting pressure rise will be smaller.

These experiments are all made with comparatively small engines, in which the action of the cylinder walls upon the steam is relatively greater than in large cylinders. They show that as the expedients for weakening wall action (superheating and jacketing) are more effective, the influence of varying compression upon steam consumption is weaker; and increase of size will have a similar effect.

Combining the indications of experiment with the reasoning under Fig. 1, it can be seen that if working conditions are such as greatly to diminish or even to abolish initial condensation, the average quality of the

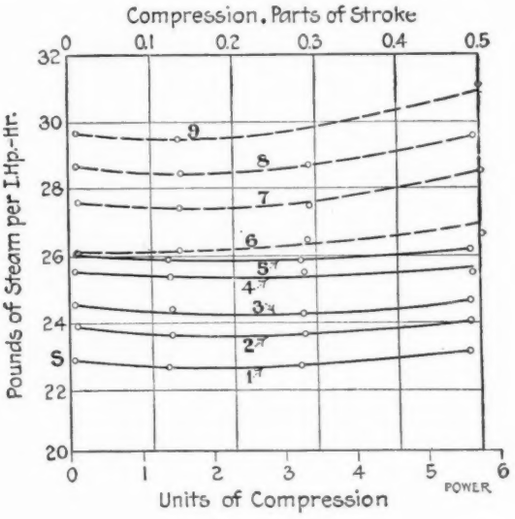


FIG. 15.

Heinrich tests, saturated steam of 116 lb. absolute pressure, cutoff kept constant at 0.18 of stroke; engine run condensing, exhaust-pipe pressure, 2.1 to 2.5 lb. absolute; clearance, 0.087; clearance surface varied; speed, 92.5 r.p.m. Curves 1 to 5, full steam jackets; curves 6 to 9, no steam jackets.

total steam during expansion may be as high as the average quality of the clearance steam during compression. With high superheat, for instance, say of 200 to 300 deg. F., the steam in the cylinder at cutoff will not have been cooled all the way down to saturation temperature because of its surrender of heat to the cylinder walls.

These considerations show that there is nothing unreasonable or contradictory to the data here set forth, in the very high economy given by the Stumpf "uniflow"

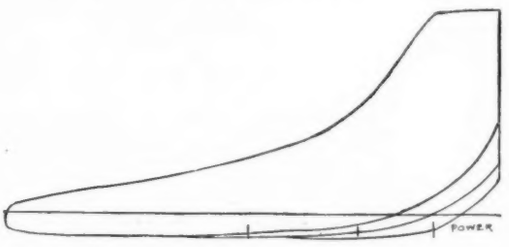


FIG. 16. INDICATOR DIAGRAMS WITH CURVE NO. 6, FIG. 15

engine, which of necessity has large clearance and full compression, nearly or quite up to admission pressure. With every care taken to diminish or prevent initial condensation, the expanding steam may have as high quality as the compressed steam. This would make a partial expansion curve such as *JK* in Fig. 1 practically run back along *FE*, thus greatly diminishing the shaded area which represents loss, and making the curve of best compression approch form *B* in Fig. 4. Under conditions of this sort, it is possible for compression to neutralize in very large degree the bad influence of clearance.

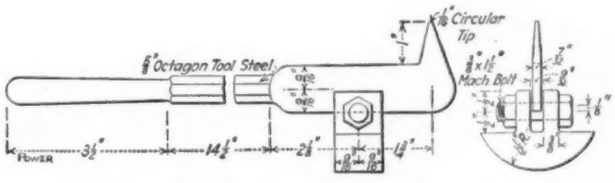
On the question of steam quality during compression—a matter which is discussed by everyone of the experimenters quoted—it is enough to say that all the data indicate that the clearance steam is by no means free from moisture at the beginning of compression, but may have a dryness fraction running as low as 0.7 to 0.5. Since superheating will exert a more direct and relatively greater effect on expansion than on compression, this helps to support the hypothesis of practically equal average steam quality for the two curves in certain high-grade engines.

Tool for Detecting Weak Boiler Tubes

BY C. A. CRAN

Engineers using Illinois coals under water-tube boilers are confronted with the problem of tube corrosion close to the tube sheets or headers, caused by the accumulation of soot and fine ashes. Our boilers are of the Heine type and are fitted with a steam soot-blower installation. This keeps the larger part of the tube surface in good condition, but it does not prevent the collection of soot and ash on the tubes for a space of about four inches just inside the tube sheet. When a boiler is shut down and cooled off, this deposit becomes damp, and the tube in contact with it corrodes away so thin at the ends as to require removal, while the balance is good for much longer service. Our chemist has shaken this soot up with water and found the water acid to litmus paper and giving tests for sulphuric acid, evidently derived from the high sulphur content of the coal used.

The writer designed the tool shown to detect tubes weakened by corrosion. To use it, the curved piece is placed



TOOL FOR DETECTING WEAK BOILER TUBES

just inside the end of the tube and the lever on it, with the sharp point against the tube surface to be tested. A pull of about 75 lb. will pierce a tube when corroded to $^3/_{64}$ in. thickness, but a tube over $\frac{1}{16}$ in. thick will give a springy or yielding feel when force is exerted on the tool so that weakness is shown before the tube is on the point of failure. The tool will make no impression on a new No. 11-gage tube. The dimensions shown are for 3½-in. tubes, but may readily be altered to suit tubes of other sizes, and the diameter of the tip may be varied to suit the ideas of the user. The lever and fulcrum are not permanently bolted together, as it is frequently easier to insert them separately.

Condenser and Heater Calculations

By Frank R. Wheeler

SYNOPSIS—Charts for readily finding the square feet of cooling surface required in a surface condenser, the amount of cooling water required, and the amount of heating surface for a heater. The derivation and use of the charts are explained.

All problems involving the transfer of heat from one body to another contain a factor called "temperature head" or pressure. In surface condenser and closed feed-water heater calculations, this temperature head is the difference between the steam temperature and a mean temperature of the water circulating through the tubes. The arithmetic mean is often used, but in these calculations, where the temperature of the out-going water so nearly approaches the steam temperature, a more accurate value is necessary. The following formula, generally attributed to Grashof, and also derived by C. P. Poole

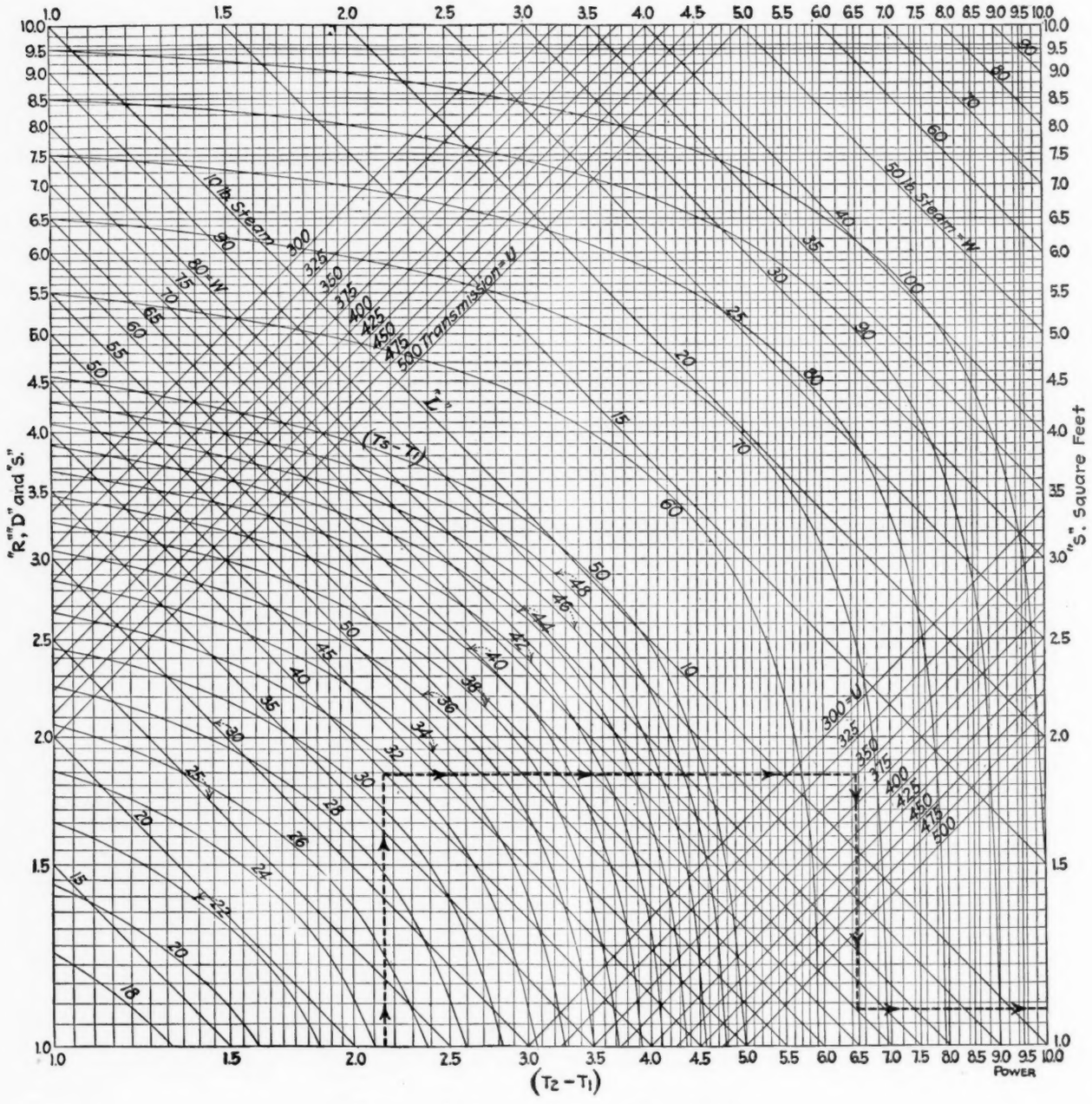


Fig. 1. Chart for Surface Condensers

$$S = \frac{W L}{U\left(\frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}}\right)}$$

$$R = \frac{L}{(T_2 - T_1)}$$

where S = square feet of tube surface; W = pounds of steam per hour; L = latent heat, asumed as 1000; U = B.t.u. transmitted per square foot per hour per degree difference of temperature, T_s = temperature of vacuum; T_2 = outlet temperature of water; T_1 = inlet temperature of water; R = ratio of condensing water to condensed steam.

(POWER, Vol. 26, p. 738), as well as Hausbrand and Lucke, is as follows:

$$D = \frac{T_2 - T_1}{log_e \frac{T_s - T_1}{T_s - T_2}}$$

Let S = Square feet of tube surface;

T_s = Temperature of steam;

T_2 = Temperature of water leaving tubes;

T_1 = Temperature of water entering tubes.

SURFACE CONDENSERS

A surface condenser is an appliance to effect a transfer of heat from steam to circulating water and its heat balance may be expressed by the formula:

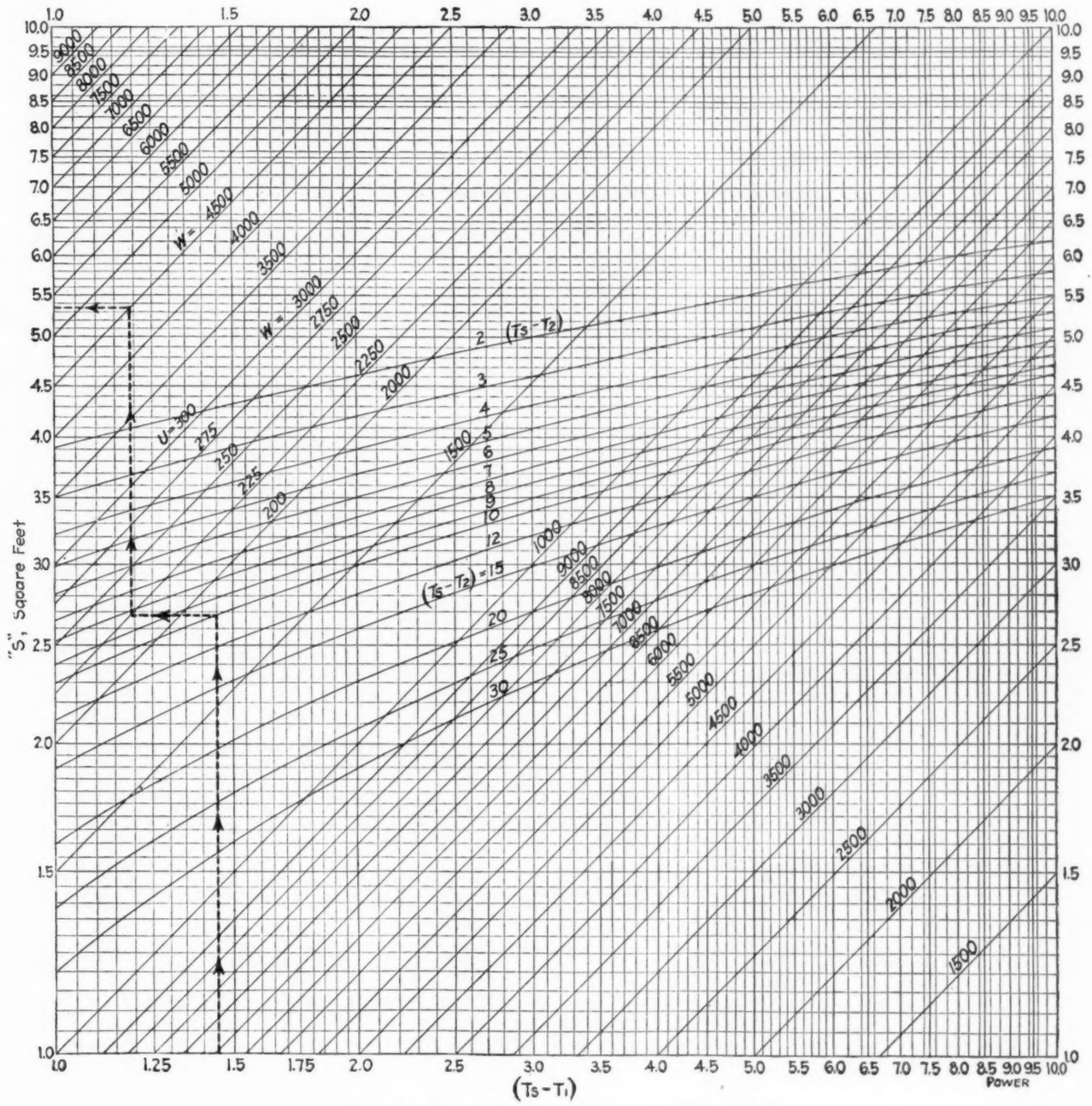


FIG. 2. CHART FOR FEED-WATER HEATERS

$$S = \frac{W\,(T_2 - T_1)}{U\left(\frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}}\right)}$$

where S = square feet of heating surface; W = pounds of water per hour; T_s = temperature of steam; T_1 = inlet temperature of water; T_2 = outlet temperature of water; U = B.t.u. transmitted per square foot per hour per degree difference of temperature.

U = B.t.u. transmitted per hour per degree difference of temperature per square foot;

L = Latent heat in steam corresponding to its temperature;

W = Pounds of steam condensed per hour;

w = Pounds of water heated per hour;

D = Mean difference in temperature between water and steam;

$$USD = WL$$

$$S = \frac{WL}{U\left(\frac{T_2 - T_1}{log_e \frac{T_s - T_1}{T_s - T_2}}\right)}$$

This is a cumbersome formula to use or plot, but the function

$$\left\{\frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}}\right\}$$

which contains three variables, may be simplified by letting $(T_2 - T_1) = a$ and $(T_s - T_1) = b$, whence

$$D = \frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{(T_s - T_1) - (T_2 - T_1)}} = \frac{a}{\log_e \frac{b}{b - a}}$$

Now assume values for a and b and solve for D, obtaining a series of curves as shown on Chart No. 1. Having found D, the formula is of simple form, which plots in straight lines on logarithmic paper.

Example 1: To condense 70,000 lb. of steam per hour and produce a 28-in. vacuum (referred to a 30-in. barometer) with circulating water at 70° F.

Assume the temperature of the water leaving the condenser to be 10 deg. cooler than the incoming steam, $T_s - 10 = T_2 = 91.5$. Let the heat transmitted per hour per degree difference of temperature per square foot be 350 B.t.u. Then,

$W = 70{,}000$;
$T_s = 101.5$ deg. F.;
$T_2 = 91.5$ deg.;
$T_1 = 70$ deg.;
$U = 350$;
$(T_s - T_1) = 31.5$ deg.;
$(T_2 - T_1) = 21.5$ deg.

Enter Chart No. 1 at 21.5 on the base, the value of $(T_2 - T_1)$ and extend vertically, as shown by the dotted line, to 31.5, the value $(T_s - T_1)$; read horizontally to 350 the value of U; vertically to 70,000, the value of W (70 being the same as 70,000 on logarithmic paper); horizontally to the edge of the chart and read 10,670, the number of square feet of cooling surface required.

In accordance with commercial practice of the condenser builders, the value L has been taken as 1000, which corresponds closely to the heat content of steam in the condenser, which, before entering the turbine, had 175 lb. pressure and 100 deg. superheat. The value L would be about 950 for steam entering the turbine at 150 lb. dry and 900 if entering the turbine dry, but at a pressure of about 5 lb. gage. Therefore, for the two latter conditions, 5 and 10 per cent., respectively, should be deducted from the final value obtained.

Circulating-Water Ratio

The actual ratio of circulating water to condensed steam is determined by the formula

$$R = \frac{f L + q - (T_c - 32)}{T_2 - T_1}$$

Where R = Pounds of circulating water per pound of steam;
f = Quality of the steam;
L = Latent heat of the steam;
q = Heat of the liquid in the steam;
T_c = Temperature of condensate;
T_2 and T_1 as before.

Due to the large difference in capacities between the different commercial pump sizes available, this formula may, for practical use, be written

$$R = \frac{L}{T_2 - T_1}$$

using the same values of L as before.

Assume the same problem and start at the base of Chart No. 1 at 21.5, the value of $(T_2 - T_1)$; follow vertically to the longest diagonal marked L (10); then horizontally to side of the chart and read 46.5, the value of R. Therefore, to condense 70,000 lb. of steam per hour and produce a vacuum of 28 in.,

$$\frac{70{,}000 \times 46.5}{8\frac{1}{3} \times 60} = 6510 \textit{ gallons per minute}$$

of circulating water at 70 deg. is required.

Feed-Water Heaters

The function of this apparatus is similar to a surface condenser and its heat balance may be expressed as follows:

$$U S D = w (T_2 - T_1)$$

or

$$S = \frac{w (T_2 - T_1)}{U \left(\frac{T_2 - T_1}{\log_e \frac{T_s - T_1}{T_s - T_2}} \right)} = \frac{\log_e \frac{T_s - T_1}{T_s - T_2}\, w}{U}$$

or

$$\frac{SU}{w} = \log_e \frac{T_s - T_1}{T_s - T_2}$$

Assuming values for $(T_s - T_1)$ and $(T_s - T_2)$, the curves shown on Chart No. 2 are obtained. Calling the value $\log_e \frac{T_s - T_2}{T_s - T_2}$, k, the formula becomes $S = \frac{K\ w}{U}$, which, as before, can be plotted as straight lines on logarithmic paper.

Example 2: To heat 45,000 lb. water per hour from 67 to 202 deg. with ample exhaust steam available at 212 deg. F. We have

$W = 45000$	*Assume* $U = 225$
$T_s = 212$	$(T_s - T_1) = 145$
$T_2 = 202$	$(T_s - T_2) = 10$
$T_1 = 67$	

Begin at 1.45 at the base of the chart, value of $(T_s - T_1)$; and read vertically to 10, the value of $(T_s - T_2)$; then horizontally to 2.25, the assumed value of U; now vertically to 45,000, the value of w, and horizontally to the edge of the chart and find 535, the number of square feet of heating surface required.

For the Engineer's "Don't Book"—Some don'ts from the book of instructions issued by the Susquehanna Coal Co. which are equally good for the engineers' don't book:

Don't forget that thorough and efficient examinations and inspections before accidents bring better results than the same work after an accident.

Don't forget to help educate the employees to take care of themselves.

Don't wait for the general inspector to point out dangerous conditions.

Don't fail to take every precaution to insure the safety of the workmen.

Don't fail to have investigated at once any dangerous conditions reported to you.

Don't employ a man for work with which he is not familiar without giving him instructions.

*When R=50, the gallons of water per minute is one-tenth of the pounds of steam condensed per hour.

Motors, Their Principles and Uses--VI

By Gordon Fox

SYNOPSIS—The concluding article of the series, taking up electrical features of motor installations, including starters, protective devices and methods of wiring.

Upon the driving equipment of a factory depends its entire productive capacity, and in order to guarantee reliability, all electrical equipment must be installed in a thorough manner. The decreased cost for maintenance will more than offset any extra costs incurred in a first-class initial installation.

Next to the motor itself, the control equipment is of first importance. This is often a point of weakness and it is possible to greatly impair both the effectiveness and the reliability of a drive by control devices not best suited to the requirements.

The principles of direct-current starting boxes and control panels were discussed in an earlier article,* although a few details deserve attention at this point. The ordinary control equipment of a direct-current motor consists of a starting box, a switch and a fuse cutout. There are on the market simple slate panels, upon which are mounted various types of starters, together with a switch and fuses or with a double-pole circuit-breaker. The circuit-breaker type is preferable, except for widely fluctuating loads or where heavy starting current is required. The panel board is compact, inexpensive and easily installed and the cost of upkeep is low.

Ordinary starters are designed for moderate duty at infrequent intervals. Where heavy loads are to be accelerated or where repeated starting is required, special resistance must be furnished. Starters of the sliding-contact type must necessarily have a large number of resistance steps in order to reduce the arcing occurring when the lever is moved from one segment to another. In the larger sizes it is difficult to prevent all arcing and also to obtain good contacts. Multiple-switch starters are preferable for heavy work, since little or no arcing occurs, and excellent contacts are secured through the use of laminated brush construction, in which case it is not necessary to provide as many resistance steps.

Drum-type controllers are commonly used where continuous or frequent control is desired, as they are more durable. They are best suited for variable-speed machine-tool work and for all dirty locations. A drum controller can be mounted readily at any point on the driven machine and, if inaccessible to the operator, a chain drive, spur gear, bevel gears, rack-and-pinion or swivel-joint device can be arranged to bring the point of control to the desired location. They are not ordinarily fitted with protective devices, but where desired, a no-voltage release and circuit-breaker can be furnished. More commonly, a fused switch is used.

Where direct-current motors are to have their speed controlled by the armature-resistance method, it is important that the resistance be properly apportioned to the load. The speed of the motor depends upon the voltage at its terminals. This, in turn, is dependent upon the resistance drop, and the latter is proportional to the current. If the resistance is not properly proportioned for all loads, then the speeds will not be correct. Consider a case in which a 10-hp. motor, rated at 1200/600 r.p.m., drives a printing press. This is approximately a constant-torque drive and the load varies directly with the speed. Assume that the motor requires 4 hp. at low speed. The resistance is designed to reduce the armature voltage from line pressure, 220 volts, to about 110 volts, so that the motor will run at half speed. The resistance drop will be correct only if the load be 5 hp. With 4 hp. there will be about 88 volts drop, and the speed will be about 750 r.p.m. The load will increase somewhat so that the speed will not be as great as this, but it will materially exceed 600 r.p.m., as desired.

This point may be made more clear by means of a set of curves, as shown in Fig. 1. Curve *A* shows the speed regulation with full voltage at the armature; curve *B*, with a small resistance inserted, and curve *D* with all the resistance inserted. It is evident that for a given load, for instance 5 hp., the speed will depend upon the resistance. Also with a large amount of resistance in the armature circuit the speed varies widely with the load. To obtain a desired speed, at a designated load, it is necessary that the resistance be of the proper value. This should illustrate the necessity of carefully predetermining load conditions for machines using this type of control and arranging the control resistance in accord with the requirements.

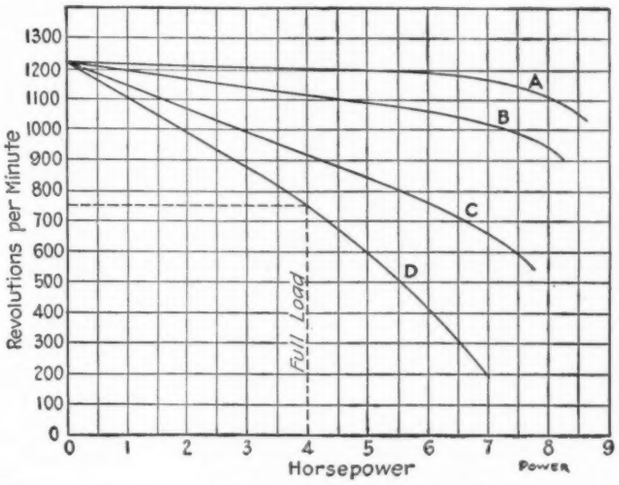


Fig. 1. Speed Load Characteristics for 0, 10, 30 and 50 per Cent. Armature Resistance

Direct-current motors having their speed controlled by the field-resistance method, have a definite speed at each controller point, changes in load having little effect. Therefore, the points above mentioned concerning armature control do not apply in the same manner. It is necessary, however, to consider the relations of power to speed to prevent seriously overloading the motor at any point in its range. For this service, controllers in which armature and field resistance are both included in the

*See "Power," page 448, Sept. 29, 1914.

same unit are preferable, since an interlocking device is provided which insures starting the motor under full field. If the motor be started with a weak field, a much heavier current is required for a given torque, since the magnetic intensity, a component of torque, is low.

Polyphase alternating-current motors up to and including 3 hp. are generally provided with a simple single-throw switch, and the fuses are made sufficiently heavy to withstand the rush of current at starting. As there is usually little inertia to these light drives, the accelerating period is short and the fuses will not blow. Five-horsepower motors are generally started by means of double-throw switches, fused on the running side only. This arrangement is also used for smaller motors where the starting load is heavy. Squirrel-cage motors larger than 5 hp., which are to accelerate under load, should be started under reduced voltage. Seven and one-half-horsepower motors driving fans or otherwise starting light, may be started on full voltage with a double-throw switch in some cases where voltage regulation is unimportant. Voltage reduction during the starting period is obtained through auto transformers, called compensators, through insertion of resistance, in the primary circuit, by taps in supply transformers and by star-delta connection of motors.

The compensator permits a selection of voltage to correspond with the starting torque required. It is efficient and reliable, and the switch is immersed in oil, thus decreasing the arcing at the contacts. Resistance starters usually have air-contact devices, but they do not reduce the starting current as effectively as compensators, due to the loss of power in the resistance. A simple starter of this type is that using variable-pressure carbon cylinders for resistance. This is quite satisfactory for motors of small size. The use of taps from the supply transformers is feasible for a few installations. Usually half-voltage taps are the only ones available. With only half voltage impressed upon the motor, its torque is low, so that this practice is confined to motors starting under light loads. The star-delta method is attained by the use of special winding connections, allowing two groupings of the stator coils, and it is necessary to carry out six leads from the motor to the switch. This arrangement also provides but one value of starting voltage, and it is adapted for light starting work only.

In selecting starting devices, attention should be given to those points calculated to render the device foolproof. In the case of compensators it should be impossible to throw the motor to running position without pausing at the starting position. It should also be impossible to leave the handle at the starting point. The interlocking devices on the best types of commercial compensators furnish this protection. No-voltage release devices are on the compensators only. Some manufacturers furnish them only as an extra. They protect the motor by throwing the compensator to the off position in case of failure of the voltage. Then, when the service is again continued, the motor must be manually started; without this protection it would receive full line pressure while at rest and serious injury might result.

Slip-ring motors are started with full voltage impressed upon the primary circuit and resistance is inserted in the secondary circuit. Inasmuch as the rotor or secondary circuit has high resistance during the accelerating period, its current is reduced and that taken from the line by transformer action is lessened. As the motor speeds up, the resistance may be cut out until full speed is reached with the controller short-circuited. Comparatively heavy starting torques can be obtained by this method with attendant low starting currents. The method is desirable for very heavy starting service. When slip-ring motors are used for variable-speed service they act similarly to direct-current motors with armature resistance. The points mentioned in that connection with regard to the relations of speed to load apply to these motors also.

Overload protection for any motor may be had through the use of fuses, circuit-breakers, overload releases and time-limit relays. Fuses have the advantage of low first cost and the disadvantages of high maintenance expense and delays for renewals. Their ability to withstand instantaneous peaks without blowing is a point of value. They are rated at maximum continuous-current carrying capacity, and will usually blow at current values about 25 per cent. above their rating, depending upon the condition and duration of the load. Table 1 shows the standard sizes of fuses interchangeable in the same clips. It is

TABLE 1. INTERCHANGEABLE FUSES FOR STANDARD CUTOUTS

Cutout Rating	Standard Fuse Sizes, Amperes
30 amperes.....	1—2—3—4—5—6—7—8—9—10—12—15—20—25—30
60 amperes.....	35—40—45—50—55—60
100 amperes.....	65—70—75—80—90—100
200 amperes.....	110—120—130—140—150—160—170—180—190—200
400 amperes.....	225—250—275—300—325—350—375—400
600 amperes.....	425—450—475—500—525—550—575—600

advantageous to have different ratings of the same size of different colors so that a casual glance will detect any substitution.

Circuit-breakers are commonly used, especially on direct-current circuits; they afford excellent protection, cause minimum delay to the service, eliminate costly fuse renewals; they can be made to break heavy currents safely and with minimum arcing; they can be arranged to signal in case of opening, but their instantaneous action renders them unsuited for some fluctuating loads.

Overload release coils are provided on some direct-current starting boxes and alternating-current compensators in place of fuses. They usually operate by short-circuiting the no-voltage release magnet. In the larger sizes of direct-current starters the no-voltage release coil actuates the overload-release device which is similar to a circuit-breaker. This arrangement is necessary to prevent destructive arcing at the first segment. In connection with alternating-current compensators, no-voltage and overload-release attachments are useful and should be used for most large sizes, due to their ability to eliminate costly fuse renewals. Definite time-limit and inverse time-limit relays, as their names imply, will carry instantaneous peaks but will operate under maintained overloads. They operate similarly to overload-release coils and are well adapted to service where fluctuating loads occur.

The wiring layout of a factory is, of course, a matter depending largely upon local conditions, the details of each installation requiring special consideration. The subject will be treated in a general way and data given to assist in making calculations.

In any factory of considerable size, it is practically compulsory that the power and lighting-distribution systems be separate; this is necessary to secure satisfactorily uniform lighting service, reliability and independ-

ence. The usual power voltages are 220 and 440, and the best lighting voltage 110, inasmuch as tungsten lamps may be used. With isolated direct-current plants, this can be accomplished with a three-wire generator and compensator. If 220-volt direct-current service only is available, it is frequently best to install a small balancer set and distribute through a three-wire system. Where alternating current is used, it is a simple matter to transform to the desired voltage. If the system is polyphase, it is best to arrange to approximately balance the lighting load on the phases if this load exceeds 10 per cent of the motor load. Two transformers or autotransformers are necessary to accomplish this result. The latter are cheaper, while the former add an insulating link in the installation.

A large plant should be divided into sections usually corresponding to departments or buildings. Each division should have its separate feeder circuit. This system gives maximum reliability through isolation of circuits, it affords centralized control, uniform voltage at all points,

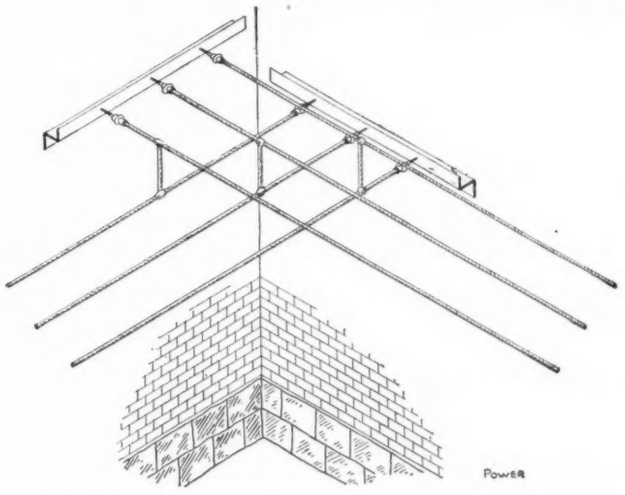


FIG. 2. ONE METHOD OF MAKING RIGHT-ANGLED TURN IN LINE

flexibility and ease of extension through the use of additional feeders and permits a check upon power consumption of departments if desired.

The most advantageous location for distribution wiring is upon the ceilings below the floors fed. Where motors are wall or ceiling mounted the distribution wiring can be readily carried upon the same ceiling. Where the installation is mixed it is often best to maintain the layout plan and possibly isolate the floors. This plan may not be necessary and, in many cases, one set of feeders may suffice for ceiling motors as well as machines on the floor above. In a single-story building with timber flooring, circuits beneath the floor may or may not be advisable, depending partly upon accessibility. Where there are concrete floors, the wiring may be in concealed conduits or tunnels. It is not desirable, however, if the machines are frequently relocated. In such cases a checkerboard connecting-in system may be arranged if the expense is warranted.

Open wiring is employed extensively for heavy work, heavy circuits being handled usually through the use of single-conductor cable. The best type of construction is that employing straight line layouts with the feeders dead ended at the extremities of a run through strain insulators. Split eyes support the cable and keep it in line but they do not clamp it. Where a turn is desired, one set of wires crosses the other at a different elevation and jumpers connect the points desired. This arrangement is shown in Fig. 2. Lighter wires may be run neatly by the use of the ordinary clamping cleats. Care should be taken to maintain good tension, support the wires at intervals not greater than 4½ ft. and do all work thoroughly. Considerable light wiring is being done by the use of multiple-conductor cable, thus lowering the work of installation materially; in many places this practice is to be commended.

Conduit work is desirable in a factory for all light work involving the use of No. 6 B. & S. wire or below. It should be used for all wiring exposed to injury, hence for most vertical runs, on posts and about machines. Conduit may also be advantageous in places exposed to moisture, acids or fumes; the ends being sealed with wax or rosin. The usual rules for conduit work should be met. For alternating-current work all wires must be in the same conduit, and for direct-current work this practice is desirable. Splices must not be pulled into the conduit.

For work involving copper heavier than No. 8, B. & S. gage, stranded cable should be used. Table 2 shows the

TABLE 2. CONDUIT SIZES, INCHES

	B. & S. Gage or Circ. Mils.	2 Wires	3 Wires	4 Wires	5 Wires	6 Wires
Solid	12	¾	¾	¾	1	1
	10	¾	¾	1	1	1¼
	8	¾	1	1	1¼	1¼
	6	1	1¼	1¼	1½	1½
	5	1¼	1¼	1¼	1½	2
	4	1¼	1¼	1½	1½	2
	3	1¼	1¼	1½	2	2
	2	1¼	1½	2	2	2½
	1	1½	2	2	2½	2½
Stranded	0	2	2	2	2½	3
	00	2	2	2½	2½	3
	000	2	2	2½	3	3
	0000	2	2½	3	3	3½
	250,000	2½	2½	3	3½	3½
	300,000	2½	3	3	3½	4
	350,000	3	3	3½	3½	4
	400,000	3	3	3½	4	4½
	500,000	3	3½	3½	4	4½

conduit sizes required for numbers of wires based on runs having not more than 4 ninety-degree turns.

Flexible metal conduit is excellent about machines for motor and controller leads. It is easily installed and fitted into out-of-the-way locations. While loom is sometimes used for this work, it is inferior.

The proper wire sizes for power distribution depend upon heating and upon resistance and reactance voltage drop. For short runs the former is the limiting feature but for long circuits of No. 0 wire or larger, carrying alternating current, the reactance drop must be considered, particularly where the wires are spaced some distance apart. Tables of values of impedance drop in various circuits are given in electrical handbooks. In circuits of large wire the reactance drop may equal or exceed the resistance drop.

The permissible voltage drop varies according to the source and cost of power, distance of transmission and character of load. In the case of an isolated plant, where the feeder system is properly laid out, it may be allowable to run the generator at a voltage as much as ten per cent. above motor rating and permit this drop in the distribution circuits. In most instances, a lower drop must be figured. The motors must have full rated voltage, or nearly so, especially if they carry overloads or require good starting torque, as these quantities vary as the square of the voltage. Theoretically, the drop should depend upon the comparative cost of power and copper.

The percentage of power loss corresponds approximately to the percentage of resistance drop. If the power loss per annum is made equal to the interest on copper per annum, the system is supposed to represent the best investment. Wherever there is a probability of future extension or a possibility of replacing a motor or motors by larger sizes, the wiring should be installed to handle the heavier service.

For the sake of facilitating estimates, Table 3 is given,

TABLE 3. APPROXIMATE AMPERES PER TERMINAL FOR MOTORS

Hp. of Motor	220 Volts			
	Dir. Current	Single-Phase	2-Phase	3-Phase
1	4.3	6.2	3	3.2
2	8	10	5	6
3	11.8	15	7.5	9
5	19	26	13	15
7½	29	40	20	22
10	37.5	50	25	29
15	50	70	35	41
20	75	..	48	55
25	95	..	54	62
35	133	..	80	93
50	188	..	110	127
75	280	..	165	192
100	373	..	215	248
200	738	..	410	475

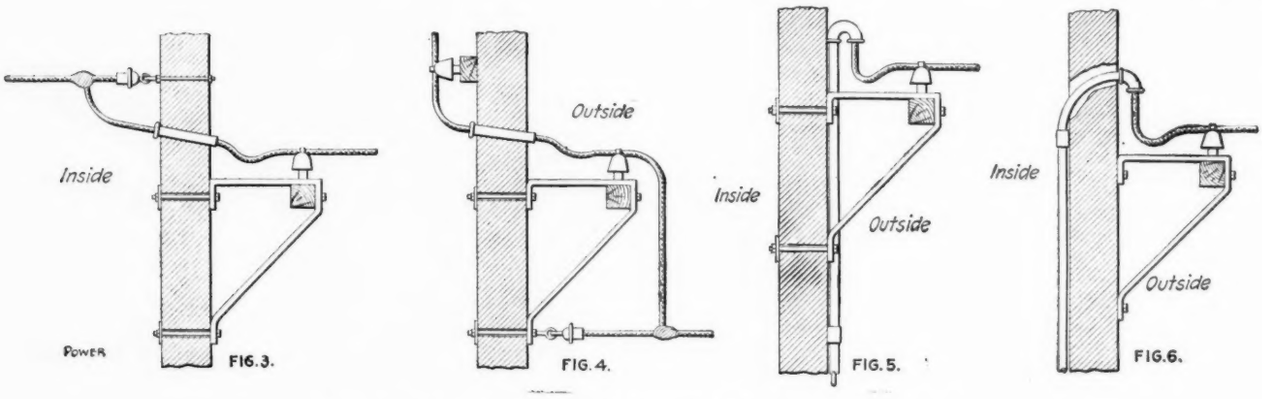
For other voltages multiply by $\frac{220}{\text{voltage}}$

on poles are best placed on platforms using two, three or four poles as legs, depending upon the weight. Care should be exercised to place the transformers advantageously, so as to save secondary copper.

An Air Pump with No Clearance

An air pump, invented by E. W. Christie, mechanical engineer of the Wheeler Condenser & Engineering Co., at Carteret, N. J., solves the question of clearance by using a central belt port (like the Unaflow engine's exhaust) for an inlet port and making the cylinder head the exhaust valve.

Referring to the accompanying section, when the piston is at the end of its stroke the central port is uncovered and the cylinder fills with air and vapor of the pressure existing in the condenser, together with some water. The first movement of the piston on the return stroke covers this port and the air and vapor are compressed and, with the water, discharged at *AA* when the head which forms a discharge valve is lifted from its seat against



FIGS. 3 TO 6. METHODS OF LEADING WIRES INTO BUILDINGS

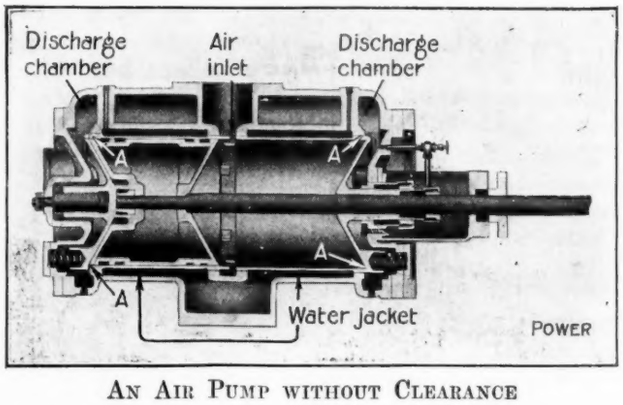
showing full-load current ratings of motors of different types. These values must be increased by 25 per cent. before figuring the wire sizes; this provides for starting currents and overloads. The Underwriters require that wiring be designed for 125 per cent. full-load current ratings and the fuses shall have a rating not higher than the listed current capacity of the wires they protect.

For plants where miscellaneously scattered points are to be fed it is often advantageous to carry most of the distribution system outside where it is more accessible and better maintained. Usually the wiring is rather heavy in character. Circuits may be carried on poles or on brackets attached to the buildings; the former practice is often preferable, but more expensive. For large cable straddle-type insulators should be used. The practice of dead-ending through strain insulators can often be advantageously followed, making a neat and substantial job. Circuits should be protected by lightning arresters at points of entering buildings. The method of entrance depends largely upon the weight of wire and type of construction; Figs. 3, 4, 5 and 6 are included by way of suggestion.

Supply transformers may be in a separate fireproof building or they may be mounted on poles. Very large transformers are best installed by the first method; the cases should then be grounded. Transformers mounted

the tension of the seating springs. The movable heads are made to fit closely into the piston so that the clearance is eliminated.

The cylinder barrel is completely water-jacketed except at the air-inlet ports. The pump is of the "wet air" type, removing both air and condensate, but may be used for



AN AIR PUMP WITHOUT CLEARANCE

air only by furnishing a slight amount of sealing water. It is adapted to either the crank-and-flywheel construction or for connection to direct-acting steam cylinders, with which, regardless of the variable stroke, the air is completely discharged and a high vacuum is thereby main-

tained. The combined equipment—direct-acting air and condensate pump at one end, steam cylinder in the center and direct-acting circulating pump at the other end—usually good for 25 to 26 in. of vacuum—will, with the Christie pump, maintain a vacuum of 28 in. or more. In tests at the shop, guaranteed performances of 28 in. with 80-deg. water are easily surpassed. A higher vacuum result was as follows:

Barometer	30.2 in. mercury
Vacuum attained	29.4 in. mercury
Pressure in condenser (abs.)	0.8 in. mercury
Temperature sealing water	60.0 deg. F.
Pressure corresponding to 60 deg. F.	0.522
Pressure in excess of that of vapor	0.278

The pressure in the condenser was less than 0.3 of an inch more than the vapor tension due to the temperature of the sealing water, showing that the pump maintains a vacuum nearly the theoretical maximum.

Chain Grates under Vertically Baffled Water-Tube Boilers*

By Osborn Monnett†

SYNOPSIS—With a properly designed arch and a restricted throat opening, it is possible to secure a smokeless setting up to 50 per cent. overload. Minimum head room for water-tube boilers of different tube pitch equipped with different types of stoker.

The vertically baffled water-tube boiler setting with a chain grate requires more careful study than the horizontally baffled setting. Cases have been known where two apparently identical settings have for some inexplicable reason shown widely different results, one being a success and the other a failure. There seems to be less certainty of smokelessness with the vertical baffle, owing to the dampening action on combustion of the bank of tubes over which the flames must pass if the flame travel is short.

For high capacities, 100 per cent. overload and above, the vertical baffle has been abandoned entirely on account of smoke trouble, and some type of the previously described tile-roof furnace substituted. For loads up to 50 per cent. above rating, a number of vertically baffled settings have been successful, and there is no question but that they will give satisfaction if properly designed.

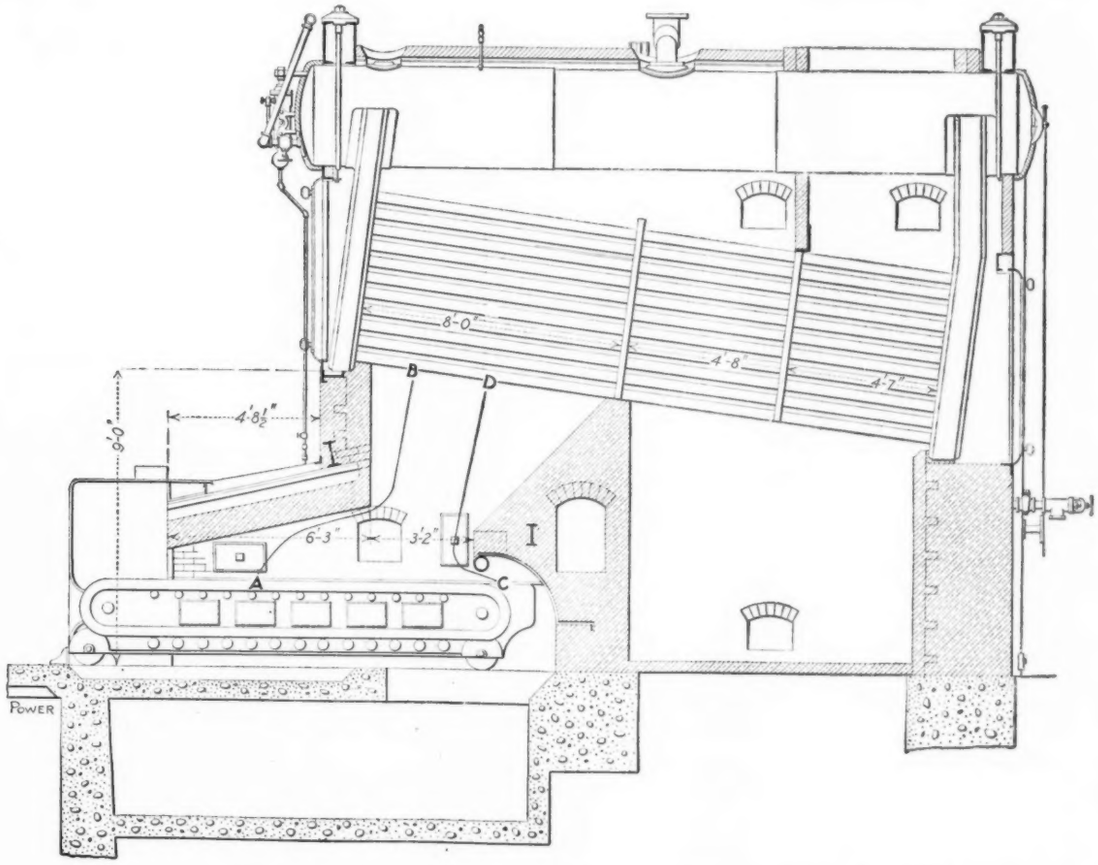


Fig. 1. Faulty Chain Grate Installation with Vertical Baffles

There is considerable objection to the chain grate among many engineers on the grounds of inefficiency. This is due to the characteristic weakness of the chain grate in allowing air to enter through the back portion of the grate and around under the water-back. Improp-

*Copyright, 1914, by Osborn Monnett.
†Smoke inspector, city of Chicago.

erly installed, the chain grate is wasteful. Properly installed, it can be made efficient. For example, take Fig. 1. As the coal passes under the arch and ignition occurs, the volatile matter begins to distill off. Following the lines of least resistance, it flows along the under side of the arch and passes into the heating surface, as shown by the line *A-B*. At the same time, air is entering through the back end of the grate and through the open-

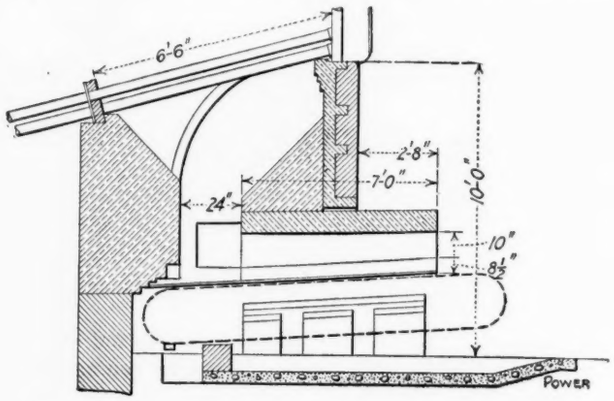


FIG. 2. B. & W. BOILER, 500 HP., WITH LACLEDE-CHRISTY CHAIN GRATE AND SPRUNG ARCH SET HORIZONTALLY

ing under the water-back, and passing into the heating surface along the line *C-D*. Thus, if the volatile matter, which will burn if supplied with air, and the necessary air to burn it are entering the furnace at points so widely separated that they do not mix, an inefficient, smoky setting is the result If the throat of the furnace is closed up to 30 in. between the bridge-wall and the end of the ignition arch, and the bridge-wall is brought up

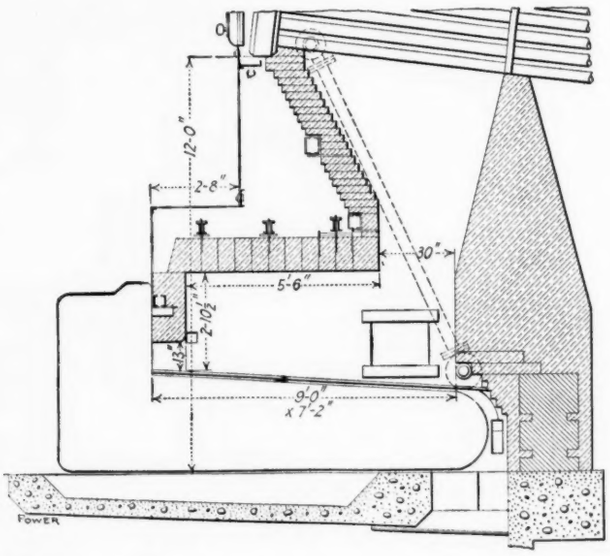


FIG. 3. HIGH FLAT ARCH WITH VERTICAL BRIDGE-WALL AND 30-IN. THROAT OPENING, 350-HP. UNIT

vertically to a point above the end of the ignition arch, the air entering around the water-back will be forced to mingle with the volatile matter in a zone of high temperature This will produce a Bunsen burner effect, resulting in a smokeless furnace and high economy. The vertical bridge-wall has a beneficial effect not only on smoke, but on the capacity to burn coal. The radiating effect of the bridge-wall helps to consume the incandescent carbon on the grates so that there is no danger, at any reasonable capacity, of live coals getting into the ashpit. The most economical way to run a chain grate is to have the grate covered and so adjusted that there will be a small amount of incandescent matter, mostly red-hot ashes, falling into the ashpit.

One of the valuable points to be brought out in Fig. 2 is the fact that it is not necessary to install a chain grate full extension with a vertically baffled boiler to get a satisfactory setting. This is important when floor space is at a premium. In this case, the extension is only 2 ft. 8½ in. to the gate, while the arch is 7 ft. long, giving a 24-in. throat opening. As a matter of fact, having the grate to some extent under the boiler enables the designer to locate the bridge-wall at a more favorable point with respect to the first baffle, and also makes it possible to get a better bridge-wall action than can be obtained with a full extension furnace. The throat area can be made small enough to insure that the volatile matter and the air from the rear will become intimately mixed in the

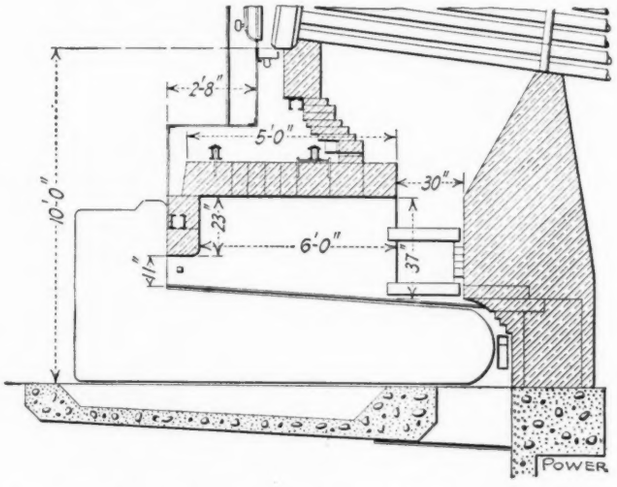


FIG. 4. ANOTHER TYPICAL FLAT-ARCH INSTALLATION WITH GOOD BRIDGE-WALL ACTION, 300-HP. UNIT

zone of high temperature. In this particular installation, the throat has been restricted more than usual, and the reduction in area has reached the limits of economy.

Another type of arch which has worked out well is shown in Fig. 3. It is installed under a 350-hp. Oil City boiler, equipped with an Illinois chain grate having a flat arch. Inside the gate there is a 12-in. arch set 13 in. above the level of the grate. Above is a 5-ft. 6-in. arch set 34½ in. over the grate, as indicated. The effort here has been made to provide unusual volume in the furnace to take care of large amounts of volatile matter. The bridge-wall is built vertical and the throat opening is 30 in. Fig. 4 shows another typical setting with this type of arch, which is good for 50 per cent. overload.

MINIMUM HEIGHTS

The height of a setting is generally measured from the floor to the bottom of the front header by engineers, architects and contractors, who invariably want to know the minimum height in which a setting will make good. For this reason, it has become necessary to fix the heights of various combinations based on experience with suc-

cessful settings. To do this, a standard set of conditions has been assumed as follows:

1. Water-back to be set 5 ft. from the inside of the front header.

2. The flame travel to be measured from a point 3 ft. in front of the water-back.

3. The distance from the latter point to the nearest heating surface to be a minimum of 7 ft.

Using the above conditions as a standard, the set of drawings shown in Fig. 5 were worked out for different stokers and tube pitches, showing the height from the floor line to the bottom of the first tube. These heights represent actual minimum conditions under which satisfactory installations may be expected, up to 50 per cent. overload, when using a successful type of arch construction, a vertical bridge-wall and a 30-in. throat opening.

It is always desirable to give dimensions in round numbers if possible, and in such shape that brick masons can work to them conveniently. From the data given, two general rules can be deduced which meet approximately the standard conditions above expressed, any variation being due to the slight difference in height of stokers and the construction of the front header in different boilers.

1. All boilers having a tube pitch of 3¼ in. to the foot, 10 ft. under the front header.

2. All boilers with a tube pitch less than 3¼ in. to the foot, 9 ft. 6 in. under the front header.

All types of arch will not be a success under the above conditions. The best ones have been the sprung arch placed horizontally, and the high, flat arch similar to that shown in Figs. 3 and 4. The step-up arch has not always worked out successfully, as it gives a poor throat opening and bridge-wall action, and the single inclined flat arch is still a doubtful proposition with these minimum dimensions on vertical baffles. Enough experience has been had with the two types first named to know that they will make good.

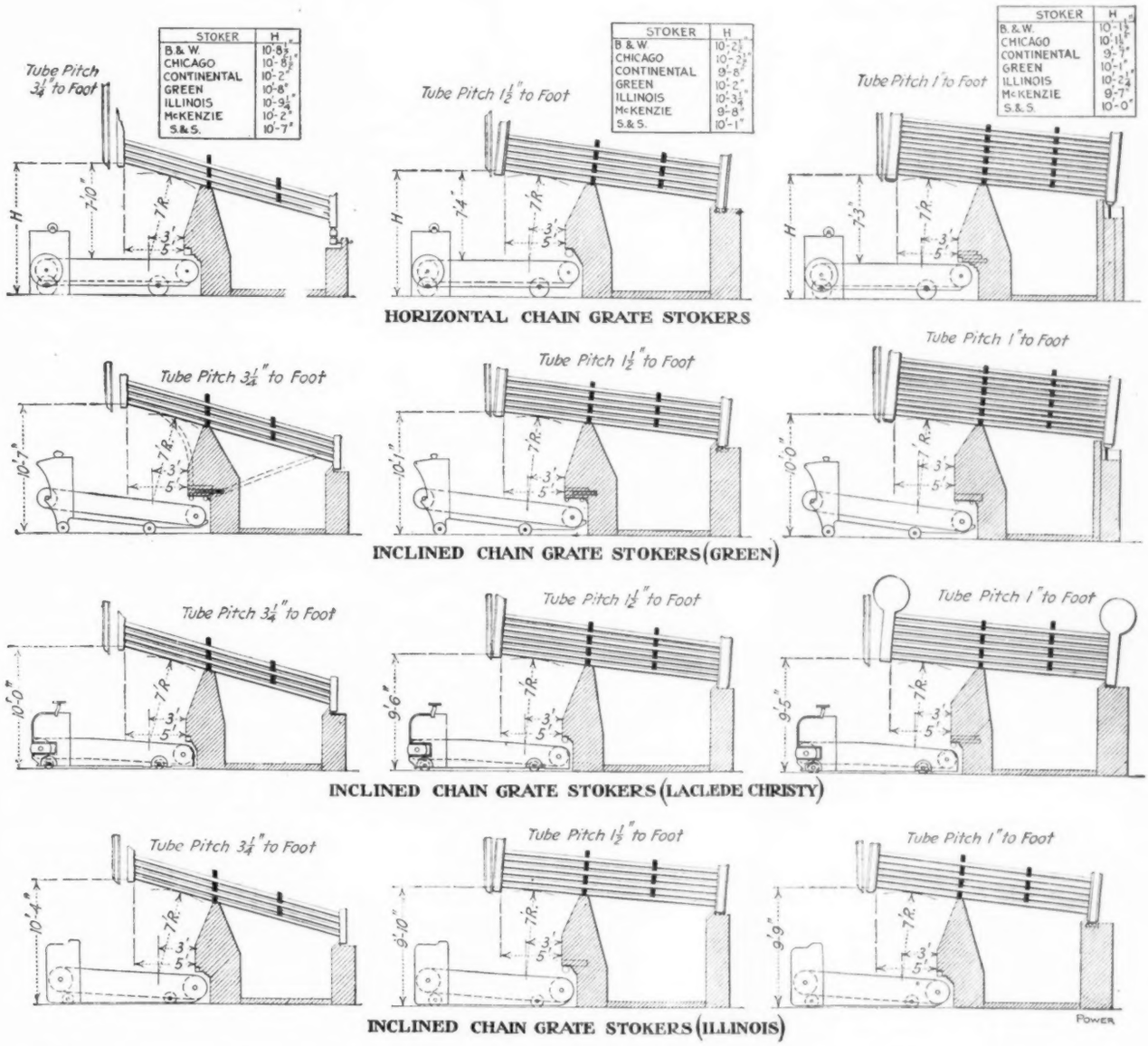

HORIZONTAL CHAIN GRATE STOKERS
INCLINED CHAIN GRATE STOKERS (GREEN)
INCLINED CHAIN GRATE STOKERS (LACLEDE CHRISTY)
INCLINED CHAIN GRATE STOKERS (ILLINOIS)
POWER

FIG. 5. STANDARD MINIMUM HEAD ROOM FOR VERTICALLY BAFFLED WATER-TUBE BOILERS AND CHAIN GRATES

Overtime Work—The engineer should keep his work well in hand so that overtime work for repairs, etc., may be avoided. Such work is usually at a higher rate and, being carried on after hours, is frequently poorly done.

"Giant" Fuel-Oil Engine

A fuel-oil engine operating on the two-stroke cycle has been placed on the market by the Chicago Pneumatic Tool Co., of Chicago, Ill. Primarily, it is intended for driving air compressors for portable and stationary service, but with regulation within 3 per cent. as claimed, it could be used for generator driving or general power purposes. It is built in sizes from 12 to 45 hp. and with twin arrangement these capacities are doubled. The fuel may be of any grade between crude oil and gasoline. The absence of valves and gearing, igniters, spark plugs, etc., results in a simple engine capable of withstanding hard usage.

The engine, which is shown in Fig. 1, has a heavy inclosed frame and in accordance with the latest designs is provided with a crosshead. The front, or crank end of the cylinder is inclosed, with a stuffing-box for the piston rod, the charge of air for the combustion chamber being compressed in this space. This air is admitted through a port uncovered by the piston at the end of the out stroke. On the return stroke, it is compressed to about 6 lb. pressure, and when the port to the combustion chamber at the top of the cylinder is uncovered by the piston, it rushes into the cylinder, pushing the exhaust gases through the port shown in Fig. 2 at the bottom of the cylinder. On the out stroke the charge of

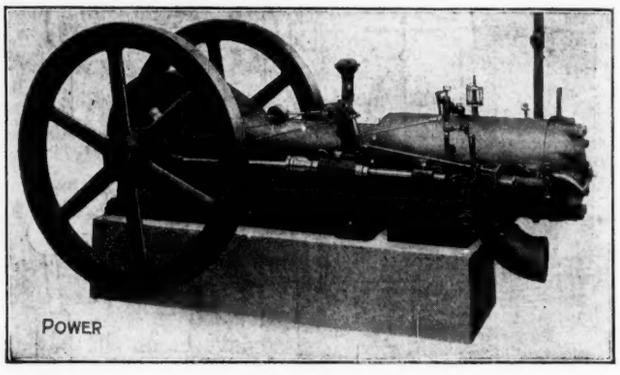


FIG. 1. "GIANT" FUEL-OIL ENGINE

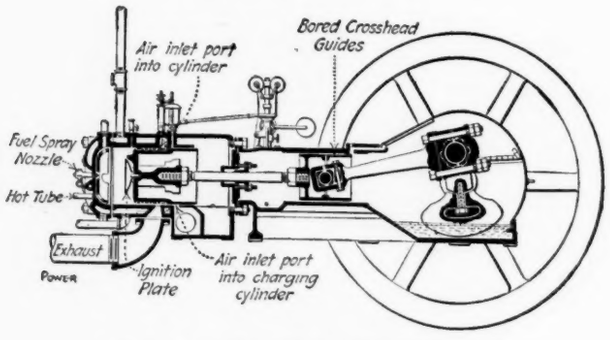


FIG. 2. SECTION THROUGH ENGINE

fresh air in the combustion chamber is compressed to about 150 lb. As the piston reaches the end of the stroke a charge of oil is injected in a fine spray onto an ignition plate attached to the piston and as it is heated during compression and by previous explosions, this plate gasifies and ignites the fuel. A hot tube is provided for starting.

The small engines may be started by hand, but for the larger sizes a small vertical single-acting air compressor, driven from a pulley bolted to the flywheel, is used. The compressor delivers air at 150-lb. pressure to a storage receiver which is of sufficient volume to run the engine on air until firing of the fuel charge begins, after which the air is shut off.

A plunger pump of simple construction, Fig. 3, supplies the oil to the engine at the center of the cylinder head through the spray nozzle shown in Fig. 4; by means of screw plugs both are easily accessible. Regulation of

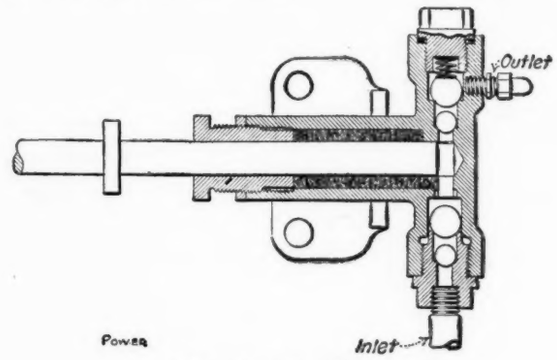


FIG. 3. OIL PUMP

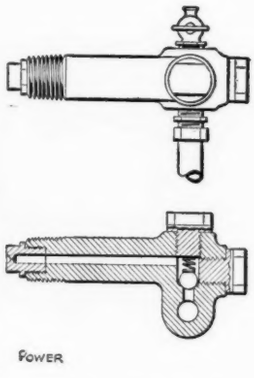


FIG. 4. OIL SPRAY NOZZLE

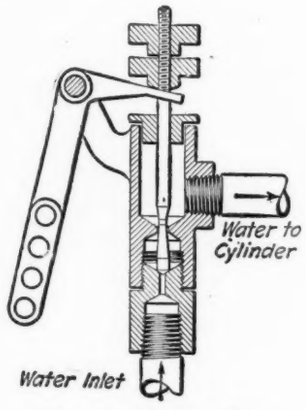


FIG. 5. WATER REGULATOR

the pump stroke is accomplished by a cam under the control of a flyball governor belted to the main shaft. The cam rests against a collar on the pump plunger, its position determining the stroke of the pump and consequently the quantity of fuel injected. A hand-operated lever also acting upon the plunger is provided for stopping the engine. The working stroke of the pump is effected by an eccentric on the main shaft through a reach rod coming in contact with the end of the plunger, which is returned by a spring; the arrangement is shown in Fig. 1.

To prevent overheating of the piston head and the burning of the lubricating oil, water is injected into the cylinder to mix with the fuel. Its supply is in proportion to the load and the quantity of fuel injected is controlled by the regulator shown in Fig. 5. The latter is also attached to the governor.

As is shown in Fig. 2, the combustion chamber and the cylinder head are water-jacketed. A sight-feed oiler lubricates the cylinder and in the crank case splash lubrication is employed. The crank is counter-balanced, and as far as the crosshead, the construction follows closely steam-engine design.

Editorials

The Jersey License Situation

On another page of this issue, A. L. Case replies to our editorial, "What's Wrong in Jersey," in the Oct. 6 issue. To many it will be gratifying to be assured by Mr. Case that Commissioner Bryant is doing all he can to complete the organization of the License Bureau. We, too, believe he is doing all he can. We believed it when we published the editorial referred to, and we further believe that the commissioner has tried to do so much that he actually got little done. The commissioner means well, but he is trying to please too many people and the old fable has it about right on the probable success of that endeavor.

We also think it was well for the commissioner to endeavor to secure the services of "the third appointee," which Mr. Case mentions. This man has successfully conducted his present plant for many years and has had experience on the engineers' license board in his own city that should be of value to the commissioner's department. But the law states that the appointees shall be approved by the civil-service commission, and when it would not approve of this one, it must have been because the commission's examiner reported that he did not pass the two examinations given him.

Notwithstanding this, this man might make a better examiner than those who will eventually be approved. But the point is this: Should the enforcement of a law that the engineers of New Jersey have worked for twenty-seven years to get enacted be allowed to drag along for so many months without better reasons than those put forth? We do not think so. There are now good reasons to believe that the necessary appointments will soon be made and that the bureau will begin its work. This is gratifying.

Specialization in the Power Plant

Few industrial workers today need to be urged to specialize in their duties. The subdivision of labor in large organizations forces many a man endowed with intiative and imagination to confine his working activities to pretty narrow areas. The power plant meets this problem in the organization of its operating staff—how should the men be classified and their responsibilities assigned? With the close figuring practiced in many big installations it is obvious that there can be no such overlapping of duties as is often found in smaller plants; little connection appears between the activities of the water tenders and those of the switchboard operators so far as the relations of the men are concerned; and the extent of supervision provided leaves little opportunity for mutual intercourse.

The exact opposite prevails in the small installation. The engineer may not shovel coal unless in a great emergency, but the variety of his duties and the interest of the boiler-room force in what goes on on the other side of the fire wall are usually much greater; a sort of interchange goes on from one end of the station to the other, and where this is encouraged by the chief without letting it get so far along as to undermine individual responsibility, it makes for highly efficient service. Generally speaking, the smaller stations call for men of the all-around type; but this by no means indicates that their work must necessarily be superficial. Which of these two diametrically opposed methods of operation is the better?

It is easy to answer that the big plant is the place for the specialist and the small one the place for the all-around man, but that is not sufficiently conclusive to be helpful. The small installation cannot afford an elaborate organization anyway, and to the great credit of the men behind the throttle and the shovel in hundreds of such plants it must be acknowledged by the most ardent advocates of extreme specialization that a great organization does not in itself guarantee the highest operating efficiency. The personal equation is the chief factor in successful service everywhere. The real answer is that a combination of the two points of view is needed to secure the best results, a combination that is unquestionably already responsible in many medium-sized installations for the excellent operating records obtained in them.

All-around individual service will continue to be demanded in the small station, and if thorough work, team play and absorbing interest in every branch of operation are cultivated little more can be asked. The greatest danger of the small plant is that the multiplicity of tasks falling upon its limited staff will tempt men often overburdened to superficial performance. Fortunately, a genuine professional pride exists to safeguard the operating man against this, and within the limits of its possibilities, the small station is doing remarkably well in innumerable cases. Its operating staff may not be known as specialists in national engineering circles, but by taking an interest in what the big plants are doing, by comparing notes with other fellows in the same range of outputs, and by doing each little job as well as it could be done by a large organization, real gains will be recorded.

As for the large station, its specialists may delve deep into the secrets of heat-unit losses, the proper adaptation of equipment to loads, the articulation of main units and auxiliaries, etc., but they cannot escape the necessity of comprehensive administration. The more that the men in every such station department can learn about the workings of other branches of plant service, the better. Let the man whose time is occupied with the analysis of coal and ashes, water and lubricants, gain at least a general knowledge of how the various generating units are handled on the floor. Let the switchboard operater familiarize himself with the path of the fuel from the barge to the grate and let the boiler room shifts be given an indicating kilowatt meter, showing the instantaneous output of the plant wherever possible. The installation of such instruments in the last year or so is one of the great-

est steps ever taken toward coalescing the interests of the men on the steam and the electrical ends of station service. No doubt, responsibilities must be pretty rigorously defined and observed in plants of immense size, but no printed set of rules is ever likely to cover the necessary procedure in every possible operating emergency, and the broader the interest, willingness to help outside exact limits and appreciation of the other man's point of view, the better. It is a question if an excessively restricted organization can be as effective in power-plant service as in the past has been believed, and anything that can be done to encourage men to look over the edges of their own little grooves and to see where all the different channels of service lead is to be commended.

Shall Railroads Inspect Their Own Boilers?

One of the points at issue between the Committee on Uniform Boiler Specifications of the American Society of Mechanical Engineers and the railways is whether the legislation which the committee proposes for universal adoption shall accept the inspection of boilers used by the railway companies by their own inspectors in lieu of by state inspectors.

In Massachusetts and the states and cities based on the Massachusetts law, the inspectors of companies doing boiler insurance must be approved by the state authorities, and when so approved become pseudo state officials, although paid by the insurance companies. They report the results of their inspections to the state department as well as to their own companies and no other inspection is required.

The proposed legislation contemplates, we believe, the inspection by the state of all boilers used which do not come under the jurisdiction of the Interstate Commerce Commission. In this way all railway boilers will be inspected either by the federal or the state authorities, and this applies to locomotive-crane, pile-driver, hoisting and stationary boilers as well as to those of regular locomotives.

Each locomotive boiler has to be inspected at frequent intervals to insure uninterrupted service as well as freedom from accidents. The railway companies employ a corps of men to keep up these inspections, and the Interstate Commerce Commission accepts their inspection of boilers used in interstate service when it is unable, by reason of the army of inspectors which it would otherwise be obliged to maintain, to have them inspected by its own men. The contention of the railroads is that the state should similarly accept the inspection of the railroad boilers by its own employees.

There is a principle involved here quite different from that in the case of the insurance company inspector. The boilers would be inspected by men who depended for favor, advancement, and even for continued employment upon the owners of the boilers they were inspecting.

The managements of some railroads would be broad enough to encourage these inspectors to expose rather than to gloss over violations of and departures from the orders of the Board of Boiler Rules. Other companies than railroads, however, would be as well entitled to have their inspection done by their own employees.

An industry with a large number of boilers could simply appoint an employee as the official boiler inspector for the company and his status with regard to holding them to a strict compliance with the law would be quite different from that of a state or insurance inspector, who could insist on what he thought necessary without prejudice to his position. If a company with a large number of boilers could do this, one with a small number could appoint its engineer or chief fireman its official inspector and the law would become a farce.

Self-inspection may do as applied to locomotives, but danger attends its too broad application.

Municipal Safety-Firstism

Portland, Ore., has stolen a march on all the other cities in the United States. It has the first Safety-First Commission, the chairman of which, Harry P. Coffin, was recently appointed by Mayor Albee. For a new thing it starts off wonderfully full grown and complete, for it has departmental organization and the selection of department heads shows discrimination. The chief of police heads the Department of Safety; a general superintendent of a railroad is head of the Department of Railroad Transportation; the chief claim agent of the local railway, light and power company has charge of the Department of Electrical Transportation; the superintendent of schools, the Department of Schools—besides which there are the Department of Industries, Department of Fire Prevention, and a Traffic Department, over which the chairman of the Commission will preside.

Details regarding the plans of campaign in the various departments, and also the posters and warning signs that will be posted, show a determination to make the Safety-First movement count in that vicinity. If it accomplishes one thing—and that is expressed in one of the slogans—it will be worth all the effort—"Stop, look, think!" These three admonitions carried out in the order indicated would avert a very large proportion of the accidents we read about every day. In fact, it is claimed that 98 per cent. of the accidents are preventable.

One thing we are sorry they omitted in the organization proposed for Portland is a Department of Boilers. We earnestly commend it to Mayor Albee and Commissioner Coffin as worth inclusion in the splendid system they have. It is a good token, however, when a city interests itself in the safety of its inhabitants, and we believe that if Portland has not already got engineers' license and boiler-inspection ordinances, the local engineers would find the present administration a good one to help them to get such laws.

It would be "bully" if someone would discover some way to reduce boiler performances under different conditions to exact figures comparable without conjecture.

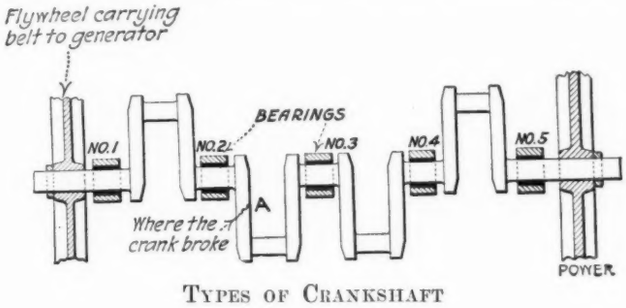
The Edison Electric Illuminating Co., of Boston, has bought from the Metropolitan Water & Sewerage Board of Massachusetts the total output of a hydro-electric plant, to be built by the state in connection with the Boston water-works system. The output of about three million kilowatt-hours annually will be furnished primarily from the discharge from a reservoir in the system. On the other hand the Edison company, together with the Boston Elevated Railways Co., will supply current to operate the motor-driven pumps in the new Boston high-pressure pumping station. Sort of a pass-it-along affair.

Correspondence

What Caused the Break?

We have 2 four-cylinder engines of the four-stroke-cycle type running on producer gas, one is a 115-hp., connected by a clutch to a 40-ton ammonia compressor, and the other, a 100-hp., is similarly connected to a 30-ton ammonia compressor and also belted to a 30-kw. generator. I came on watch at 7 p.m., and at 7:30, the crankshaft on the 100-hp. engine broke.

There was a 30-ampere load on the machine at the time. Both engines were running on the ice machines, which had a back pressure of 16 lb. and a head pressure of 180 lb. with the expansion valve almost wide open. Suddenly the belt that runs the generator almost



Types of Crankshaft

jumped off and there was an unusual noise in the crank case. I shut the engine down and found that the crankshaft was broken. When we removed the shaft, which is 6 in. in diameter, we found nearly all the babbitt wiped out of bearings No. 1 and 3, but the other three were in good condition. I should like to have opinions expressed as to what caused the crankshaft to break.

E. M. Merrill.

Chicago, Ill.

Motor-Trouble Diagnosis

Some years ago I was hurriedly sent to a machine shop to find out why an electric motor, the main drive of the shop, would not run.

Upon closing the main switch the pilot lamp burned brightly, so I proceeded to bring the starting-box lever over. As the field circuit was made, the lamp dimmed. going entirely out when the starting resistance had been cut out of the armature circuit. This performance invited a ready deduction, which the following will serve to demonstrate:

Referring to the diagram, let A represent one of the leads which has parted in the manhole, the ends being submerged as shown; C the pilot lamp; D the motor field and E the armature. Assuming a resistance of 20 ohms through the fault, and with the starting-box lever in the off position, only the fault and the lamp would be in the circuit. The lamp, being of the ordinary carbon type (110-volt, 16-cp.), would require about one-half an ampere, thereby, in accordance with Ohm's law, causing a 10-volt drop at the fault and giving at the pilot-lamp terminals $115 - 10 = 105$ volts decrease from the normal voltage of the lamp of 5 volts, which would not be very noticeable in daylight.

Neglecting the paralleling of the motor field with the lamp, their joint conductivity being relatively negligible when the motor armature is paralleled with them, we have, assuming an armature resistance of 0.25 ohm, $20 + 0.25 = 20.25$ ohms as a total resistance. This re-

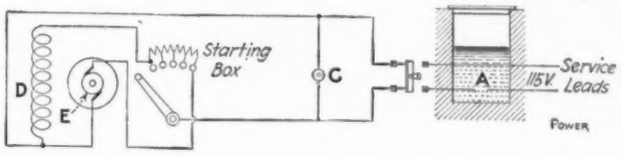


Diagram Showing Location of Fault

sistance, assuming a line voltage of 115 volts, would allow a current of $115 \div 20.25 = 5.679$ amperes.

This would cause a drop in voltage at the fault of 113.58 volts, which would leave of the 115 volts only $115 - 113.58 = 1.42$ volts, at the motor terminals.

Amos J. Carr.

Fort McKinley, Maine.

Poor Firemen

Someone of Power's editorial writers must be very unfortunate in the company he keeps, or he would not have written that editorial "Attention of Firemen" (Sept. 15). The statement that, "Seldom is any disposition shown by the men in the fireroom to study to improve their condition," is rather startling. It would be interesting to learn how many of the engineers who read Power once handled the coal scoop. Probably the percentage would be quite large. And they have not risen without some "study to improve their condition."

There comes to my mind a man working in a fireroom twenty-five miles from Boston. For the last three winters he has attended an evening technical school in that city, making the trip three or four nights a week. Other evenings, Sundays and holidays were taken up by study. This may be an extreme case. Not many men in any walk of life have "sand" enough for that.

Among my acquaintances in the fireroom are many men who little resemble the picture drawn in the editorial. If the writer of that editorial will come with me, I will take him into firerooms all day long; and in all of them, we will find men who not only can talk about evaporation, but who actually watch the log sheet with interest. In not a few, we will talk with firemen who can converse intelligently about the relation of draft to thickness of fire, and the treatment best for certain grades of coal. And unless said writer is expert in use of CO_2 apparatus and interpretation of the records, I would suggest that he refresh up a bit before we start, for we shall meet men who know a thing or two about combustion.

Occasionally, one finds a plant where the firemen "know

little and care less." In such cases, the most likely place to look for the cause of this trouble is in the vicinity of the chief engineer's chair.

Speaking of "Filling up the furnace clear to the door," brings to mind a plant of which I once took charge. That was the plan of firing there. For a few days I watched those men stand two at a furnace and heave coal until it poured back out of the doors. When I stepped in and changed things, there was a tickled bunch of firemen in that plant. They had been doing as they did because they had been made to do so. "It was killing us," they told me. All were interested in the resulting economy.

Here are some of the things that a chief engineer may do to help the firemen. In the advertising pages of POWER he will find instructive illustrations of apparatus around the plant. Get a lot of picture frames and mount each in a separate frame, adding what explanation (typewritten or printed) seems necessary. If POWER does not supply the required help, a catalog probably will. If not, a blueprint may be made.

Go out into the fireroom and show the men how you want the work done. This will interest and encourage them.

Have a standard wherever you can—a standard water level, a standard method of cleaning fires, a standard thickness of fire, a standard method of coaling, etc. Things will run smoother in the plant if you do. And hold the men to these standards—not so rigidly as to be foolish, but with wise firmness. If a man thinks he has an improvement, encourage him to come and talk it over. If it looks good, let him try it; and watch the trial. If it does not, try to show him your reason for refusal.

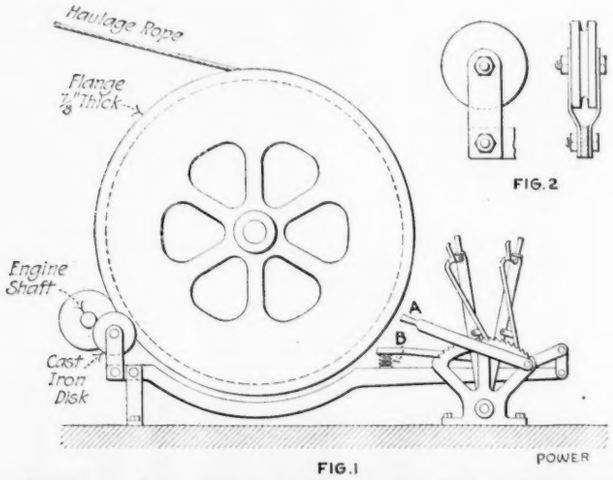
Lastly, remember that they belong to the same division of the animal kingdom as yourself.

WILLIAM E. DIXON.

Cambridge, Mass.

Reversing a Cable Drum

We often want to pay out rope on a second-motion friction-clutch haulage system having nonreversing engines. To put on reversing eccentrics and links would be quite expensive and as the engine has to run at a high rate of speed, they would be objectionable on that score also. We put on the device shown and can now pay out rope at will without reversing the engine.



FRICTION DISK BETWEEN DRUM AND ENGINE SHAFT

Fig. 1 shows the drum, the cast-iron disk, and the control levers. Fig. 2 is the cast-iron disk.

By releasing the brake and clutch and pulling up on lever *A*, the disk is pressed against the engine shaft and the flange of the drum (in the groove shown in Fig. 2), and causes the drum to rotate in the unwinding direction. *B* is a coil spring which holds the lever up and keeps the disk from rubbing the engine shaft or flange when not in use.

J. J. PIPER.

Colorado Springs, Colo.

Improving a Pump Stuffing-Box

On a large pump the stuffing-box was so shallow that it would hold but two rings of packing at a time. As they had to be kept tight, they scored the rods. While making general repairs the stuffing-box was turned out $\frac{3}{16}$ in. larger in diameter. A cast cup was then made to fit inside, and 3 in. longer, which besides reducing the size of packing required, made additional room for two more rings. A gasket was then put in the bottom of the stuffing-box and the cup placed in position and drawn up tight with nuts on the gland studs. The gland was then turned down and the job was complete and satisfactory.

LOUIS B. CARL.

Marshfield, Wis.

Sizes of Motors Driving Centrifugal Pumps

We have just read the article by E. M. Marshall on "Sizes of Motors Driving Centrifugal Pumps," which appeared in the Sept. 15 issue. Attention is called to trouble which occurs owing to disregard of pipe friction when figuring total head. The more publicity given to this

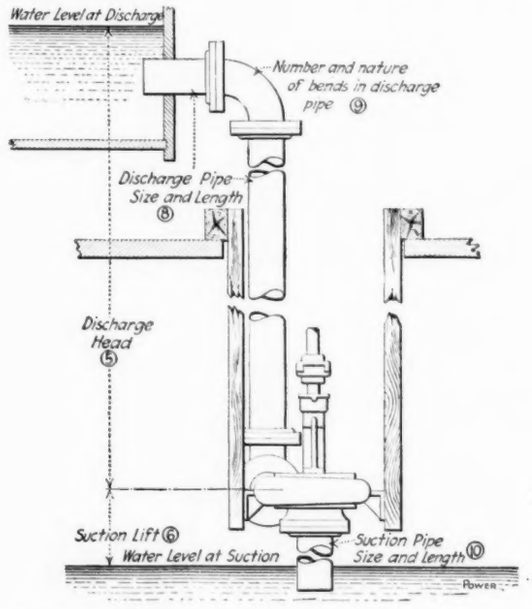

Water Level at Discharge
Number and nature of bends in discharge pipe ⑨
Discharge Pipe Size and Length ⑧
Discharge Head ⑤
Suction Lift ⑥
Water Level at Suction
Suction Pipe Size and Length ⑩
Power

FIG. 1. MEASUREMENTS FOR VERTICAL PUMP

matter the better, but it seems that the author has criticized the pump manufacturer in too general terms, particularly in the following statement.

"Manufacturers sell pumps to be connected to certain types of motors and when electrical troubles come up, they

are glad to refer them to the maker of the motor, but are little interested in securing and tabulating for future use the nature of the trouble and the remedy."

The pump manufacturer is often called upon to assume the account for both the pump and the motor, and we believe that the majority of pump manufacturers pay prompt attention to complaints in regard to motors driving their apparatus and that the motor manufacturers in turn, when such complaints are put up to them, are only too glad to investigate.

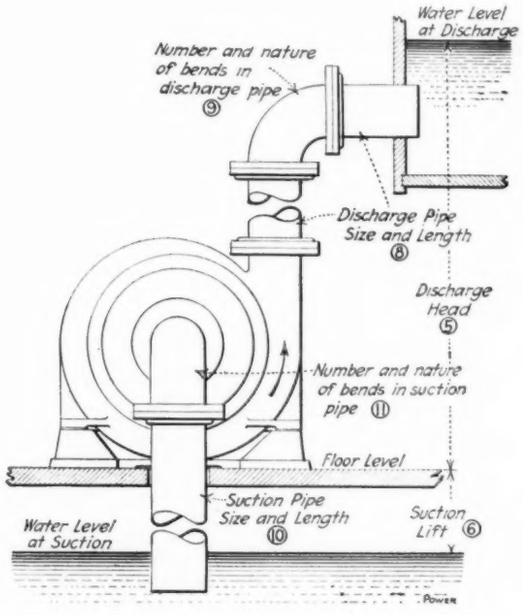

Water Level at Discharge
Number and nature of bends in discharge pipe ⑨
Discharge Pipe Size and Length ⑧
Discharge Head ⑤
Number and nature of bends in suction pipe ⑪
Floor Level
Suction Pipe Size and Length ⑩
Suction Lift ⑥
Water Level at Suction
POWER

FIG. 2. MEASUREMENTS FOR HORIZONTAL PUMP

The various motor companies do a large resale business with pump builders and naturally have to give good service. One particular reason why the motor builders like to sell their apparatus to the pump builders is that the latter usually exert great care in seeing that the proper motor is applied for the conditions under which the pump is to operate.

It is our practice to have the purchaser fill out the "Centrifugal Pump Data Sheet," herewith shown, before we start actual construction, provided the pump is to be directly connected to an electric motor. This enables us to carefully check the total head against which the pump is to operate.

CENTRIFUGAL PUMP DATA SHEET

Shop Order No.....

1. Capacity of pump.....U. S. gallons per minute.
2. Character of fluid.....salt, fresh, clean or dirty.....
3. Temperature of the liquid.....°F. Specific gravity.....
4. If solid matter is contained, state size and character of largest pieces .....
.....
5. Discharge head.....ft. (This is the vertical distance from the floor level of the pump to the point at which the water is discharged.)
6. Suction lift.....ft. (This is the vertical distance from the floor level of the pump down to the level of water in pit from which it is pumped.) (See No. 7.)
7. Suction head.....ft. (Sometimes the pump is placed so that the water will flow to it and in that case the pressure at the inlet to the pump must be given. Question No. 6 may then be omitted.)
8. Discharge pipe diameter.....length.....
9. Number of bends in discharge pipe.....State whether elbows are 45 or 90 degrees.....and of long or short radius .....
10. Suction pipe diameter.....length.....
11. Number of turns in suction pipe.....State whether elbows are 45 or 90 degrees.....and if long or short radius.....

Signed .....

Date..... Address .....

Mr. Marshall refers particularly to the greater power required due to increase of head caused by pipe friction. We would point out that the question of speed required for this greater head is likewise important in the case of a centrifugal pump.

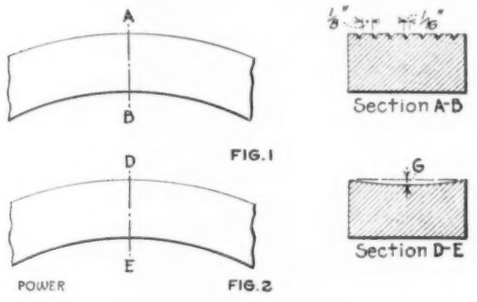
E. C. WAYNE,
Goulds Mfg. Co.

Seneca Falls, N. Y.

Machining Piston Rings

Sometimes new piston rings in steam and gas engines leak, due to a poor fit in the cylinder, and it takes a long time for them to wear to a tight joint, even if both rings and cylinder are ground to a good fit when cold.

Figs. 1 and 2 show two methods of machining the faces of rings which will cause them to adapt themselves to the cylinder bore in a very short time. Rings turned with V-shaped grooves about $\frac{1}{16}$ in. wide, spaced about $\frac{1}{8}$ in. apart, as in Fig. 1, or made slightly concave, as at *G*, Fig. 2, will soon wear down to a good running fit.



TWO FORMS SUGGESTED

M. E. GRIFFIN.

Franklin, Penn.

What Is Wrong in Jersey?

In reply to an editorial appearing in the Oct. 6 issue, I wish to rectify some of the statements made regarding the organization of the Engineers and Boiler Operators' License Bureau in New Jersey.

Having been closely connected with this matter while it was before the state legislature, and since that time, particularly during the past three months, I wish to assure the engineers interested that the Commissioner of Labor has made all reasonable effort to perfect the organization of this bureau. There is no question of truth regarding his inability to do so during the first year of the existence of this law, and the matter was taken up by him actively immediately following the appropriation made by the last legislature for this purpose.

The conditions of the law imposed a situation which has been found difficult to meet satisfactorily. These are, that the salary offered by the state for this work was no inducement to the class of men who would be desired by the department for the work, thereby eliminating most qualified men from even a desire to enter the field, and as the law also required an experience of ten years actually in charge of a steam plant of 250 hp., or more, developed another process of elimination, so that in the final determination there have been very few from which to choose, and, in addition to these difficulties, the Civil Service Commission has seen fit to adopt a rigid examination

before issuing a certificate to anyone whom the Commissioner of Labor may have selected; and if, as you express it, there is a deep feeling of dissatisfaction among the engineers, it cannot be otherwise than their lack of knowledge of the conditions which have operated to prevent the completion of this Board, and any charge that the Commissioner of Labor has played politics is without just foundation.

The facts are that one of the earlier appointees personally eliminated himself from the Board by withdrawing his name from the appointment. As the third appointee has always been considered one of the best engineers in the state, was so considered by the department, due to his known experience and business ability, it was considered but justice to him, and to the best interest of the department that extra effort be made to secure his acceptance by the Civil Service Commission, and it has only been during the present week that this matter has been finally settled. I wish to assure all interested that no attempt has been made by the Commissioner or anyone else to railroad, as you term it, anyone either into or out of office, and further that only the greatest care and investigation have been used by this department in an endeavor to appoint competent men.

I hope all interested will bear patiently with this department until it can be finally organized and accept the work of the Board as the final determination of the honest effort which has been made to make the department a working success.

A. L. Case.

Trenton, N. J.

Operating Engineer as a Purchasing Agent

The editorial with the above title in the issue of Sept. 22 is extremely interesting, and the subject matter of vital importance. Correct buying involves a multitude of economic and engineering considerations, and the one "best buy" for any specific condition is not easily found. Many purchasing agents would do well to acquire some engineering knowledge, and it seems equally true that most operating engineers sadly need grounding in the fundamentals of buying.

It is pointed out that the man higher up will listen to the neat-appearing salesman with forceful arguments much more readily than to the operating engineer in greasy overalls and a mediocre "line of talk." True, but the overalls need not be any hindrance. A man in a shipshape rig, with a clean shave, a clear, steady eye and a carefully garnered supply of specific detailed information can easily convince the purchasing agent of just what his plant requires long before the salesman has registered at the local hotel.

In visiting the man behind "the door marked private," do not forget your economics. It is a poor argument to dwell on the fact that the equipment you are recommending will pay for itself in two years or so. The same money, if invested in another department, might double itself in half the time. The man you are talking to is there to invest to the best possible advantage, and you must get his viewpoint to succeed. Have figures all worked out, and do not forget such fundamentals as interest on the investment, depreciation and all burden charges. In fact, try and imagine yourself in the owner's shoes, try and look next Saturday's payroll between the eyes, the lean sales for last month, the show-me-or-get-out board of directors, and then, ask yourself if the proposed expenditure is absolutely necessary.

Before making the visit to the front office, put your proposition down in writing. You are going to ask another man to spend his money. It is a serious business. Make sure in your own mind that you know what you want to say and how to say it. Arrange your arugments in logical sequence; talk dollars and cents first, and technical "reasons why" only if necessary, and not to display your vast engineering knowledge. When the boss thinks in terms of the balance at the bank, it is well to forget the back pressure resulting from B.t.u. lingering somewhere between the exhaust ports, and the condenser. Observing all the foregoing suggestions, take up again the matter of economy, and see if you cannot save your salary by installing the right apparatus for reclaiming unnecessary waste.

E. Hurst.

Montour Falls, N. Y.

Duplex Pumps Not Good for Jet Condensers

The question is often asked, "Why is the duplex pump not more frequently used in connection with jet-condensing apparatus?" It is frequently used as a circulating pump with surface or barometric condensers, but with jet types, where it removes the air as well as the condensing water, it cannot be used with any degree of satisfaction. This is because the steam valves are operated from the opposite side, consequently if one side should become air-bound, the work of this cylinder being less, it would operate too quickly and by so doing would shift the valve of the other cylinder in the same manner, causing a very intermittent action. This, however, only occurs when there is a considerable quantity of air present.

John F. Hurst.

Louisville, Ky.

Patching Boilers

I have just noticed the letter by Dale S. Miles on p. 355, Sept. 8, under the above caption. It is, of course, possible to derive a rule for the efficiency of a diagonal joint that will hold for extreme cases as well as for those encountered in practice, but such a rule involves the use of trigonometry and is therefore not as convenient as that given by Mr. Mason in his book, "The Arithmetic of the Steam Boiler."

Any slight inaccuracy due to the use of the rule in question, for practical boiler work, will be found to be less than that of some other factors, such as measurement, calculation and workmanship. Hence the slight difference in results by using this rule may be overlooked.

It is a fact that many practical rules in engineering do not give absolutely correct results or do not hold in extreme cases, yet these rules should not be condemned on that account.

W. G. Hawley.

Scranton, Penn.

Inquiries of General Interest

Placing Globe Valve on Feed Pipe—When a globe valve is used on a boiler feed pipe on which side of the valve disk should the inlet be?

E. J. M.

The inlet should be under the disk of the valve, i.e., tending to open the valve, otherwise should the disk become detached, it would obstruct the passage of feed water to the boiler by acting like a reversed check valve.

Standard Cubic Foot of Gas—What is meant by a standard cubic foot of gas?

G. L. C.

A cubic foot of gas taken under standard conditions of temperature and pressure, i.e., at a pressure of 29.92 in. of mercury and 32 deg. F. or the equivalent, viz.: at a pressure of 30 in. of mercury and 62 deg. F., these pressures at the corresponding temperatures being equivalent to 14.7 lb. per square inch.

Flash, Burning and Chill Points of Oil—How are the flash, burning and chill points of an oil determined?

L. F.

The flash point is found by placing the bulb of a thermometer in a porcelain dish filled with the oil to be tested, then heating the oil over a Bunsen burner and noting the temperature at which a flash is obtained from passing a lighted taper over the surface of the oil. The "burning point" is found by carrying the heating to a temperature at which the flash causes the oil to burn. The "chill point" is found by first chilling a small quantity of the oil in a test tube surrounded by a freezing mixture until the oil congeals, then stirring the congealed oil with a thermometer and noting the temperature when the oil has warmed sufficiently to be flowed from one end of the test tube to the other.

Stayed Area of Boiler Head—What is the rule for finding the area of that portion of a return-tubular boiler head which should be stayed?

F. W. A.

The area which requires staying is a circular segment 3 in. inside of the shell and 2 in. from the tubes. The formula for finding the area of such a segment is:

$$A = \frac{4\,H^2}{3}\sqrt{\frac{2\,R}{H} - 0.608}$$

in which

A = Area of segment in square inches;
H = Distance in inches from the tubes to the top of the shell, minus 5 in.;
R = Radius of boiler in inches, minus 3 in.

Mean Effective Pressure with ¼ Cutoff—With initial pressure at 100-lb. gage, cutoff at ¼ of the stroke, cylinder clearance 5 per cent., and average back pressure 4 lb. above atmosphere, what mean effective pressure would be realized?

J. G. W.

The formula generally employed for determining the average forward pressure per pound of absolute initial pressure is that for isothermal expansion, viz:

$$P_m = (1 + \log_e R)\,(f + c) - c,$$

in which

P_m = Mean forward pressure per pound of initial absolute pressure;
$\log_e R$ = Hyperbolic logarithm of the ratio of expansion;
f = Per cent. of stroke complete at cutoff;
c = Per cent. of clearance.

Multiplying the mean forward pressure per pound of initial absolute by the actual initial gives the average forward pressure, and deducting the average back pressure gives the m.e.p.

Instead of computing the average forward pressure per pound it is usually taken from tables such as those in Low's "Steam Engine Indicator," pages 115 and 116. For ¼ cutoff and 5 per cent. clearance the tables give $P_m = 0.6258$. The initial pressure being 100 + 14.7 = 114.7 lb. absolute, the average forward pressure would be

$$0.6258 \times 114.7 = 71.78 \text{ lb. absolute,}$$

and the average back pressure being

$$4 + 14.7 = 18.7 \text{ lb. absolute,}$$

then

$$71.78 - 18.7 = 53.08 \text{ lb m.e.p.}$$

Delivery per Revolution of a Duplex Pump—Allowing 5 per cent. slippage, how many gallons of water would be delivered per revolution by a duplex pump having water cylinders 14 in. in diameter by 16 in. stroke and piston rods 2½ in. in diameter?

F. G.

Each piston 14 in. in diameter would have an area of

$$14 \times 14 \times 0.7854 = 153.94 \text{ sq.in.,}$$

and for one stroke in the head end of each cylinder the piston displacement would be

$$153.94 \times 16 = 2463.04 \text{ cu.in.}$$

With piston rods 2½ in. in diameter the cross-sectional area of each rod would be

$$2\tfrac{1}{2} \times 2\tfrac{1}{2} \times 0.7854 = 4.91 \text{ sq.in.,}$$

and the net area of the rod side of each piston would be

$$153.94 - 4.91 = 149.03 \text{ sq.in.,}$$

hence for one stroke in the rod end of each cylinder the piston displacement would be

$$149.03 \times 16 = 2384.48 \text{ cu.in.}$$

Therefore the piston displacement per revolution in one cylinder would be

$$2463.04 + 2384.48 = 4847.52 \text{ cu.in.,}$$

and for both cylinders it would be twice as much, i.e., 9695.04 cu.in., or a total piston displacement per revolution of

$$\frac{9695.04}{231} = 41.97 \text{ gal. per rev.}$$

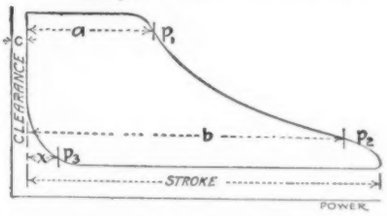
With 5 per cent. slippage there would be an actual delivery of $41.97 \times (1 - 0.05) = 39.87$ gal. per rev.

Steam Consumption from Indicator Diagram—What is the formula for computing the steam consumption of a simple engine from the indicator diagram?

W. S. K.

The formula is:

$$Q = \frac{13{,}750}{\text{m.e.p.}}\left[(F + c)\,W_f - (x + c)\,W_x\right]$$



STEAM CONSUMPTION FROM INDICATOR DIAGRAM

where

c = Clearance in fraction of the stroke;
F = Fraction of stroke completed at a point chosen on the expansion line (which point should be like p_1, near the point of cutoff, where a would represent the fraction of stroke; or p_2 near the point of release, where b would represent the fraction of stroke);
x = Fraction of stroke uncompleted at a point like p_3 chosen on the compression line;
W_f = Weight per cu.ft. of steam at pressure shown on the expansion line at the fraction a or b of the stroke;
W_x = Weight per cu.ft. of steam at the pressure of compression shown on the compression line at the uncompleted fraction of stroke x;
Q = Steam accounted for per hp. per hour.

The result will always be less than the usual amount of steam supplied to the engine, because a considerable proportion of such steam is condensed upon its entrance to the cylinder, and is not reëvaporated until after the valve opens for exhaust, so that it does not show at all on the diagram.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—EDITOR.]

Engineers' Study Course

Problems in Power Plant Design--II

POWER FOR DRIVING MACHINERY

After having made a general study of the conditions to be met, and having prepared an outline of the methods to be followed, as given in the previous article, the next step is to find the capacity of engine and generator for driving the machinery of the plant. In the machine shop there is a total of 250 hp. to be supplied to the machines, when all are running at one time. Of this, 150 hp. is to be furnished by two motors belted to the line shafting. In cases of this kind it is customary to assume that the friction of the main shafting and various countershafts makes up about 30 per cent. of the total load. Therefore, in the present case,

$$150 \div 0.7 = 214 \ hp.$$

must be delivered to the line shafting by the two motors. Assuming an efficiency of 90 per cent. for these at full load

$$214 \div 0.9 = 238 \ hp.$$

of electrical energy must be delivered to these motors.

In addition to the above, 100 hp. of mechanical energy is to be delivered to individual machines by direct-connected motors. The efficiency of small motors working under these conditions will not average much more than 60 per cent., so that

$$100 \div 0.6 = 167 \ hp.$$

must be provided for this part of the work. This makes a total of

$$238 + 167 = 405 \ hp.$$

of electrical energy to be supplied to the building when all of the machinery is in operation.

Taking up the other buildings to which power is to be supplied in like manner, the following is obtained: In the foundry, 15 hp. delivered by line shafting calls for

$$\frac{15}{0.7 \times 0.9} = 24 \ hp.$$

and 15 hp. by direct-driven motor calls for

$$\frac{15}{0.6} = 25 \ hp.$$

making a total of

$$24 + 25 = 49 \ hp.$$

of electrical energy to be delivered to the building.

In the forge shop 15 hp. delivered by line shafting calls for

$$\frac{15}{0.7 \times 0.9} = 24 \ hp.$$

and 10 hp. for direct-drive motors, requires

$$\frac{10}{0.6} = 17 \ hp.$$

making a total of

$$24 + 17 = 41 \ hp.$$

for the building.

In the pattern shop 20 hp. delivered by line shafting calls for

$$\frac{20}{0.7 \times 0.9} = 32 \ hp.$$

In the carpenter shop 5 hp. delivered by line shafting calls for

$$\frac{5}{0.7 \times 0.8} = 9 \ hp.$$

The total power required for the plant with all machinery in operation (exclusive of ventilating fans and electric pumps) is

$$405 + 49 + 41 + 32 + 9 = 536 \ hp.$$

Under actual working conditions only a part of the machines will be running at one time, and it is usually considered ample to provide 75 per cent. of the power as computed above, which in the present case, is

$$536 \times 0.75 = 402 \ hp.$$

To determine the total power load it will be necessary to estimate the power required for operating the various ventilating fans and for pumping the water from the river to the reservoir in the yard. These computations will be made accurately later on, when this part of the equipment is designed, but for present purposes an approximation of the power required for this work will be sufficiently accurate. Four changes of air per hour in the machine shop, pattern shop and toolroom, and six changes in offices and drafting room will call for the delivery of approximately 6,340,000 cu.ft. of air per hour by the fans, while the conditions of ventilation called for in the foundry and forge shop will require the moving of about 1,600,000 cu.ft. more, or a total of

$$\frac{6,340,000 + 1,600,000}{60} = 132,333 \ cu.ft. \ per \ min.$$

Under the conditions found in this class of work it will require approximately 1 hp. delivered at the fans for each 1700 cu.ft. of air moved per min., or in the present case

$$132,333 \div 1700 = 77.8 = 78 \ hp.$$

Assuming motor efficiencies of 85 per cent. This part of the work will call for

$$78 \div 0.85 = 92 \ hp.$$

or for a round number, say 90 hp.

The arrangement proposed in regard to the division of power, and for condensing, is as follows: The main engine is to be of sufficient size to drive the machinery, including ventilating fans and electric pump. It is to be run noncondensing during the winter and the exhaust utilized in the heating system. In the summer, when the exhaust would be wasted, the engine is to be run condensing to secure greater economy in steam consumption. If computations show the exhaust from the power engine to be sufficient for heating purposes, then the condensing apparatus may be used in connec-

tion with the lighting engines during the heating season, at which time the lighting requirements will be at a maximum.

If, however, the exhaust from the power engine is not ample for heating purposes, then sufficient exhaust from the lighting engines may be used to make up the deficiency while they are in use.

During the nonheating season, the power engine will be run condensing. As the nonheating and nonlighting seasons practically coincide, the lighting engines will not be used to any extent during the summer, except on cloudy days and to furnish lights at a few special machines. This leaves the condensing apparatus free for independent use by the power engine. There may be a short time in spring and fall when the lighting engines will be used with the heating plant shut down, but with compound engines having a good steam economy, it would hardly be worth while to provide condensing equipment for use during these short periods. The best arrangement would probably be to furnish a condenser of sufficient capacity for the power engine. Then run this engine condensing during such part of the year as the increased economy was found to exceed that due to utilizing exhaust in the heating system. For the remainder of the year, the power engine would be run noncondensing, and the condenser would be available for one or more of the lighting engines whenever it was found advisable to use it.

As the weight of condensing water required is from 30 to 35 times that of the steam condensed, it is evident that a supply computed on this basis will be ample for all purposes at any time of the year. For example, during the winter, with the exhaust used in the heating system and the condensation returned to the boilers, there will be a comparatively small amount of "make-up" water required, and if part, or even all, of the lighting engines are run condensing, the water requirements will probably not exceed those for which the plant is regularly designed. If, however, a larger quantity should be required temporarily, the surplus stored in the reservoir may be drawn upon to make up the deficiency.

In the summer, with the power engine running condensing, the maximum quantity of water will be required, but at this time the boilers may be fed from the cooling water after it has been used in the condenser, so that no extra amount will be required for this purpose. It will, therefore, be seen from the above that a water-supply based on that necessary for condensing in the summer time will be ample for all purposes.

Size of Main Engine

The electrical energy to be delivered to the motors in the different buildings was found to be

$$402 + 90 = 492\ hp.$$

Taking the transmission or line loss as 5 per cent., the efficiencies of generator and engine as 90 per cent. each, the indicated horsepower of engine must be

$$\frac{492}{0.95 \times 0.9 \times 0.9} = 640$$

Before computing the water required for condensing we must assume the power required for pumping this amount from the river to the reservoir, and make a trial computation. Taking this as 30 i.hp. at the main engine, it will call for a total of

$$640 + 30 = 670\ i.hp.$$

A compound condensing engine of the moderate-speed four-valve type will require approximately 18 lb. of steam per indicated horsepower per hour, or

$$18 \times 35 = 630\ lb.$$

of condensing water. For the above engine this amounts to

$$\frac{670 \times 630}{60} = 7035\ lb.\ per\ min.$$

The theoretical power required for pumping water is given by the formula:

$$horsepower = \frac{W \times H}{33{,}000}$$

in which W equals the weight of water pumped per minute and H the height to which it is raised, including the friction head. Applying this to the present case, assuming the friction head to amount to 5 ft., gives

$$horsepower = \frac{7035 \times (50 + 5)}{33{,}000} = 11.7$$

or practically 12 hp. Assuming a line loss of 5 per cent., pump efficiency 65 per cent., motor efficiency 85 per cent., and generator and engine efficiencies of 90 per cent. each, as before, the indicated horsepower to be provided in the main engine for this purpose will be

$$\frac{12}{0.95 \times 0.65 \times 0.85 \times 0.9 \times 0.9} = 28$$

which comes within the limit assumed.

Having found the required power of the engine, the next step is to decide upon the type and general dimensions. As it operates under practically uniform conditions the year around, and is of large power, the steam economy is important. Taking everything into consideration without going too much into detail, it would seem that a cross-compound engine of the medium-speed four-valve type would be well adapted to the present case. Ample room for an engine of this type is easily provided, the power house being new, and it may be direct-connected to the generator, which is a mechanical feature of considerable importance. As previously stated, it will be advisable to run noncondensing for a considerable portion of the heating season to utilize the exhaust steam, but the engine should be provided with a condenser for summer use. Engines of this type and power will vary somewhat with the make. The following, taken from the catalog of a standard company of engine builders, gives data for an engine of approximately this power operated noncondensing under a steam pressure of 20 lb.

Diameter high-pressure cylinder, in.	24
Diameter low-pressure cylinder, in.	38
Stroke, in.	27
R.p.m.	150
Floor space, direct-connected type, ft.	24×21

For power work where moderate- and variable-speed motors are desirable, direct current is required, and in this case a generator furnishing a current of this kind at 250 volts will be used. As

$$670 \times (0.9 \times 0.9) = 543\ hp.$$

of electrical energy are to be delivered by the machines, its rating should be

$$\frac{543 \times 746}{1000} = 405\ kw.$$

or 450 kw. to be on the safe side. By making the capacity this much larger than called for, it will have no appreciable effect upon the efficiency under normal working conditions, and will allow a certain surplus for additional calls for power or for lighting in the summer, without starting up the regular lighting plant. For example, on cloudy days in the early fall it will probably be necessary to run a part of the lights for an hour or so at night and a good generator should be able to carry 20 to 25 per cent. overload for this length of time without injury or serious loss of efficiency. In the present case, this would give a surplus of

$$(450 \times 1.2) - 450 = 90 \ kw.$$

for purposes of this kind.

Operating the engine condensing at the same initial pressure and cutoff, will increase the power approximately the same amount (20%) over that when running noncondensing. Hence, in the summer, there will be approximately 90-kw. capacity available in the main generating set for temporary lighting, should occasion require. The size of engine noted above has a sufficient capacity, above that called for, to drive a 450-kw. generator without overloading.

It is customary, in practice, to increase the power of the main engine or generating set about 10 per cent., to care for any moderate increase in the required power due to the installation of additional machinery. In the present case, this is provided for by using slightly larger machines than called for in order to make use of stock sizes.

Test for the Carbonizing Effect of Gas-Engine Oil

By C. L. Cook*

Much has been heard of the so-called "noncarbonizing" lubricating oils for gas and gasoline engines. This property is an important one because the collection of carbon around the valves and in the interior of the engine soon interferes not only with its efficiency but with its actual operation. In an endeavor to find some means of measuring this carbonizing effect for the purpose of passing upon various samples of oil, I have developed the following test. It is my opinion that all oils carbonize, that there is no such thing as a noncarbonizing oil, and that there is a practical limit as regards the amount of carbonization under which an oil will be satisfactory and over which it will be unsatisfactory for use in a gas engine.

Carbonizing is caused by the cementing or binding together of small particles of soot, etc., in the interior of the engine by a sticky residue left by the partial distillation of the lubricating oil when it is exposed to the heat of the burning gas. The temperature is not high enough to completely distill it, hence it accumulates, gathering more carbon, and forming an extremely hard shell or crust upon the walls of the cylinder and the valve surfaces. If the lubricating oil contains considerable gum, the engine carbonizes rapidly while if it contains a relatively small amount the engine seems to be able to get rid of it at the start and a crust is formed only after a relatively long period of running.

*Chemist, Board of Public Works, San Francisco.

The following test produces this formation of carbon from a given sample of oil and measures it quantitatively:

Exactly five grams of the oil to be tested are put into a platinum dish, and two cubic centimeters of benzine or petroleum ether are added. If less ether is added combustion may not be complete and experience has shown that more ether is unnecessary; therefore, for uniform results, this measured quantity should be used.

The dish is then placed in a cylindrical inclosure, made of a can about 6 in. in diameter and about a foot high, without a top and having holes punched in the sides at the bottom to cause a slight draft, and with a wire shelf about 4 in. down from the top, so that the dish can be put on it and the ether ignited by passing a flame over the top of the can. When the combustion has been completed, the dish and contents are weighed. At this stage the dish will be found to be coated with soot and to have a certain amount of a sticky residue in the bottom, looking like tar or asphaltum.

Next, the dish is passed over the flame of a bunsen burner until without further ignition or combustion this tarry residue is evaporated to dryness, assuming a flaky graphitic appearance. The dish and its contents should be weighed again and the loss over the previous weight is that of the gum evaporated. This is a measure of the gumming or carbonizing qualities of the oil, and can be expressed as a percentage of the five grams used for the test.

In numerous oils that were tested this percentage of gum-forming ingredients was found to vary from 0.3 to 3 per cent. For gasoline-engine oil, as used in automobile engines, this percentage should not exceed 1 per cent. One oil in particular was tested and it was found that serious carbonizing which occurred at a percentage of 1.4 per cent. stopped when this was lowered to 0.8 per cent by the use of a better compounder or base.

Lighter oils show a lower percentage of gum, whereas heavier oils have a greater percentage of the compounder which is added to give the necessary viscosity. This gum is introduced into the oil, it seems, as impurities of the compounder; hence the percentage will depend upon the amount and the purity of the compounder used. From this it is evident that the purity of the compounder to be used for a heavy oil is of the greatest importance. Estimating the amount of compounder used in a heavy oil, the writer is of the opinion that this percentage of gum-forming ingredients should not exceed 1.5 per cent. for so-called heavy oils.

The test described can be easily made in fifteen minutes. After having used it for over a year along with the usual tests for gravity, flashpoint, viscosity, and in some instances tests for acidity, alkalinity, and breaking down at 400 deg. F., the writer is now in a position to select a suitable oil at the best price and to be able to successfully interpret the almost unexplainable "kicks" which came from reliable sources and which at first seemed imaginary.

It is to be observed that this test is an arbitrary one, and is not exactly analogous to what happens in the engine cylinder; the gum as measured may or may not be a true measure of the amount deposited in the engine cylinder. If, however, the same conditions are observed in all tests made, the method becomes available for selecting a desirable oil.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

More power to George Chandler, and all engineers like him! Over 50 years he's been a stationary engineer, 17 of which he was chief engineer with the Bryant Paper Co., Kalamazoo, Mich. Now, at 74, he has voluntarily retired and the company, in recognition of his long service and loyalty, will continue to pay him his full salary. And bully for the company, too. Both boss and chief may well be proud of themselves.

"College doesn't make fools, it merely develops them," said "Old Gorgon Graham," in his letters. So, if you meet a college feller who you believe is a fool, remember he had a big start in the fool business long before he ever entered college.

Out in Portland, Ore., they have a Safety First Commission, and its apparently complete organization makes it look like Portland will soon be a mighty safe place to move around in. Man named Coffin is the chairman—no, he's a live one.

Some day we are going to cast a form for use in newspaper offices when reporting a boiler explosion. Here are some expressions that the censor will delete: "Escaped in a miraculous manner"; "hurled this mass of metal"; "it went up in the air straight as an arrow, carrying the steel roof with it"; "it blew out with terrific force"; "windows were shattered"; "concussion from the explosion";; "cause was low water"—and somewhere in the form will be this line: "If you can't find out the reason, don't fake one."

Pretty clever wording that, "Electricity the heart beat of modern warfare." But, "Peace hath her victories no less renowned than war." So let us hope that the plowshare of peace will soon drive a deep furrow through these now bloody fields of Europe and that the sun will shine down once more upon those desolated homes and bring up a harvest of happiness and prosperity. Speed the day!

To increase the use of its current, a Kansas City lighting company had demonstration days, one for washing, one for ironing, and one "when the more delicate dishes were given prominence." What's indelicate about washing and ironing? Oh, Mama, may I go out to swim?

Elevators were known as far back as 300 B. C., we were told by a wise geek at New York's electrical show. Yes, but that city is rapidly replacing them. By 2001 this type of lift will be only a memory.

Uncle Sam's navy has keen competition in its midst in the individual coaling of its ships. The dreadnaught "Arkansas" has made a record of stowing 560 tons in its bunkers in 60 minutes.

But the "Kronprinzessin Cecilie" had one true heart aboard, writes a visitor who saw her shortly after her capture, and here comes a fine touch of humanity. After making the round of the ship, our chief engineer found in the cabin assigned him a blotting-pad with a farewell message. There were none of the German jeers you might expect, but penciled appeals in broken English to clean the engines "twice weekly," to feed the "pidgons" on the "uper deck," and to "oel" the piston-rods regularly for the very good reason that "paking is cast-iren and will be rosting." It seems that the Ober-Ingenieur who wrote this note had superintended the building of the ship in the Stettin yards, and when he took leave of her at Falmouth, they say he sobbed like a woman bereft of a child. Doubtless in the Babylonian captivity of Bodmin he wonders how the engines that he loved are faring, and it should comfort him to know that they are in good hands. Moreover, his blotting-pad message now adorns a British officer's album as so much proof that a true McAndrews may bear a German name.

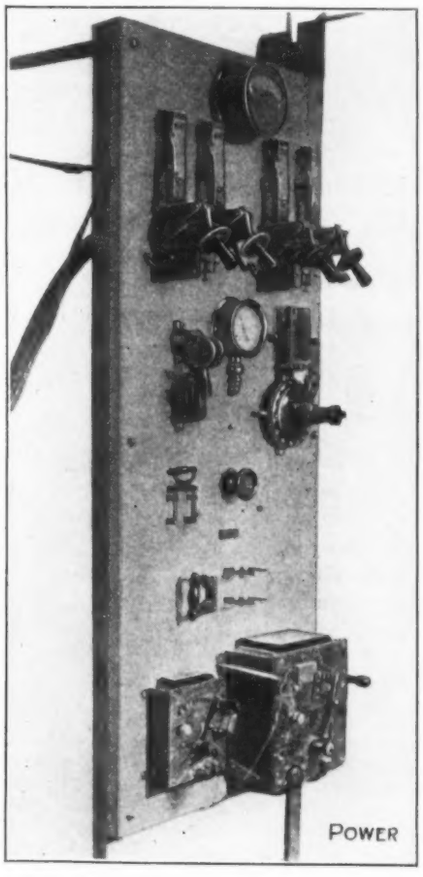
Brother Harris, Lachine, Que., sent us this corking good heart-throb from the "Montreal Star."

Electrical Control of Isolated Vacuum Heating Apparatus

By H. G. Gibson

The power plant with which the writer is connected takes care of several buildings. One of them, recently constructed, is some distance from the power house. The heating and ventilating plant in this building consists of a double-duct forced-air system, the air being heated by exhaust steam, aided by live steam in very cold weather. The steam coils are operated on the vacuum system, the condensation being removed by means of a motor-driven centrifugal pump. The air-circulating fan is also motor-driven.

Because of the isolated position of the building with relation to the main engine room, it is not possible for the



Switchboard with Pump, Fan and Vacuum Control

engineer to remain continually with the heating apparatus. It is obvious that should the vacuum pump stop for any reason, the coils would soon fill with water. If the outside temperature were below freezing and the air-circulating fan continued to run after the vacuum pump stopped, probably some frozen and burst coils would be the result.

To lessen the chance of such an occurrence the following arrangements were made: On the switchboard, with the rest of the pump and fan-controlling apparatus, was mounted a diaphragm circuit-breaker, shown at the center in the illustration. This and the vacuum gage, in the center of the board, are piped in parallel to the suction

side of the vacuum pump. When the pump maintains a vacuum in the coils, it also acts on the diaphragm switch (just above the spring), closing it. The closure completes a circuit through the coil of a solenoid-controlled, quick-acting circuit-breaker, shown to the left of the vacuum gage, this breaker being in the air-circulating fan-motor circuit.

Thus it will be seen that if for any reason the vacuum pump stops, or for any other reason a vacuum is not maintained, the diaphragm spring will open the switch to which it is connected. This breaks the circuit in the solenoid, allowing its contacts to open, thus breaking the circulating fan-motor circuit and stopping the fan.

Part of a signal apparatus for notifying the engineer when the fan has stopped is shown at the left of the air-fan starting box near the foot of the switchboard. It consists of a piece of fiber, to which two brass flexible contact strips are attached. This is so fastened to the switchboard that when the fan stops and the starting-box arm falls back, it will cause the two contacts to touch and complete a battery bell circuit to ring a bell in the engine room.

Small Hydro-Electric Plant Damaged by Flood

By C. V. Hull

The floods of the Cedar, Little Cedar and Shellrock rivers of central Iowa were unusually destructive this year, causing considerable property damage, and the dam at Marble Rock, on the Shellrock River, was seriously damaged.

The dam, and the small but uptodate power house, were completed in the fall of 1913; the only unsatisfactory part of the installation was the foundation for the dam. A view of the generating apparatus is shown in Fig. 1.

When plans were made for the plant, explorations of the river bottom showed that it was some 40 ft. to bedrock. It seemed inadvisable to build the dam on rock and consequently the power house and the dam were built on a foundation of piles with a cement cap.

When the high water came, the unusual pressure and current caused erosion and movement of the river bed, so that water passed under the dam. When this was noted, the flood gates were immediately opened wide, but despite this relief the water continued to carry away the dirt and piles until the volume of water passing under the dam was greatly increased. Probably 15 or 20 ft. of the foundation was washed out, but the dam was so well reinforced that it stayed in position with part of the foundation gone. Fig. 2 shows the water at its highest stage.

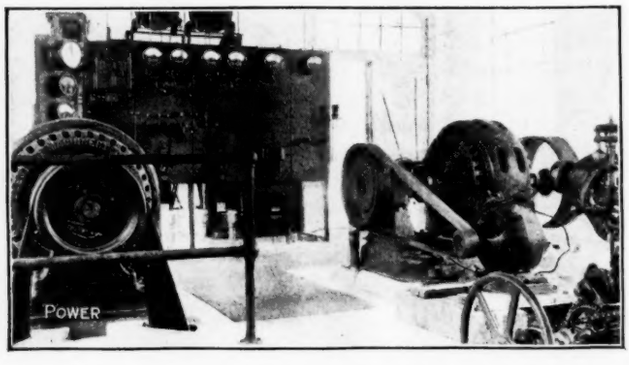


Fig. 1. View of the Station Apparatus

Fig. 2. Exterior of the Power Plant; Water at Highest Stage

The damage done to the plant will make it necessary to put about $15,000 into the repair work, and a new bridge will cost about $20,000 to replace the one carried away.

While the water power was out of service, a kerosene tractor outfit furnished power to run one of the alternators, which was turned at right angles to its former position, the belt being run through the door of the building, Fig. 3. This makeshift gives service from 14 to 16 hours per day.

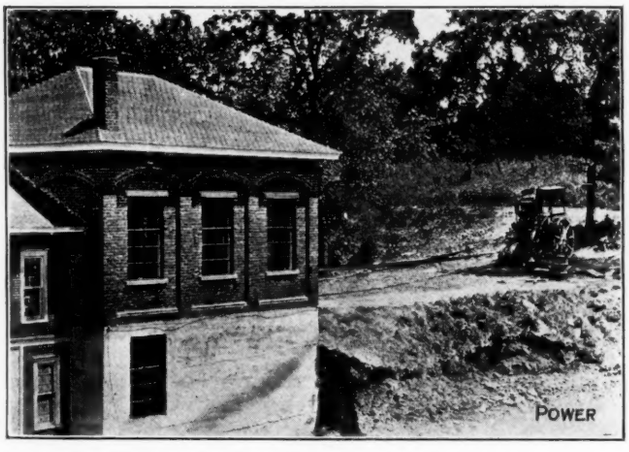


Fig. 3. Tractor Engine Belted to Generating Unit through Door

As soon as the dam can be repaired, the original equipment will again be put in service. In it are two water, vertical-shaft turbines, belt-connected to three-phase, 60-cycle, 2400-volt alternators, running at 400 r.p.m. The generators and turbines are arranged for parallel operation. The switchboard is neatly arranged, with all the necessary switches and instruments. Current is furnished to the consumer at 104 volts. The exciters run at 1550 r.p.m. and generate 28 amperes at 125 volts.

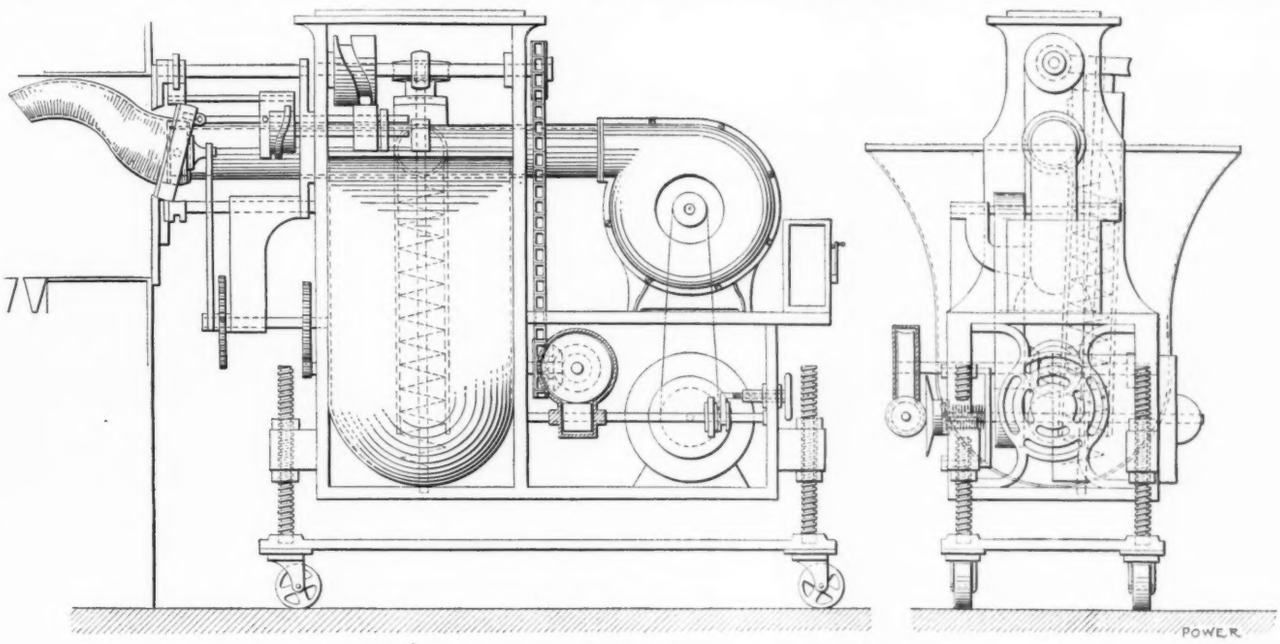
Since the turbines have vertical shafts, quarter-turn idlers are necessary to make the belt turn to the generators. They are secured to heavy frames, molded into the heavy cement walls of the buildings; this insures a substantial and accurate lineup.

The plant was put in for the benefit of residents of the town, who wanted day-and-night service.

Cooke Mechanical Stoker

A mechanical stoker embodying new features is illustrated herewith. It is attached to the boiler front by simply opening the door and running the discharge nozzle into the opening. The stoker is run by a ¼- to 1-horsepower motor, according to the size of the boiler. The motor is placed underneath the blower, the latter being driven by a belt from the former; the pipe which conveys the coal to the oscillating nozzle is attached to the blower.

The hopper is in the middle of the machine, the top of which stands about 40 in. above the floor. Coal is lifted out of the hopper by a screw conveyor into the side of the pipe and is then carried by the force of the air from the blower onto the fire. The coal and air in passing through the nozzle, absorb heat and the fuel is deposited on the fire hot; at the same time the nozzle itself is prevented from becoming overheated by the passing of this coal and air through it.



Details of the Cooke Mechanical Stoker

The coal is bituminous slack, and the stoker will operate on any coal that will pass through a ¾-in. mesh screen. By the mixture of oxygen with the coal, the finest of the coal or the dust is aflame immediately upon leaving the nozzle, and the larger particles are burst open by the heat, liberating the gases, which are immediately consumed, with the result that there is an incandescent fire in the furnace. The feed of fuel and the air supply are under perfect control, and the faster the feed, the hotter the fire.

Being adaptable to all sizes of boilers and mounted on wheels, the stoker is easily moved up to or away from the boiler. The hopper will hold about 350 lb. of coal.

The stoker is manufactured by W. H. Cooke, 325 Granite Bldg., Rochester, N. Y.

Wastes Around Power Plants—Wastes around the power plant may be reduced somewhat by the covering of all hot surfaces, such as small air-separating and other tanks which receive the condensation from vacuum-return systems, or from drips, etc. All drips should be covered when the condensation from the drips is not wasted to the sewer.

Annual Meeting of the A. S. M. E.

The annual meeting of the American Society of Mechanical Engineers will be held at the home of the society, the Engineering Societies Building, 29 West Thirty-ninth St., New York City, Dec. 1 to 4.

It is expected that two features of the annual meeting alone will make it distinctive and of widespread interest. The first of these is the important all-day session on Thursday, Dec. 3, on the Engineer in Public Service, when papers upon municipal engineering and related topics will be presented. Contributions have been arranged for upon a wide variety of subjects as is indicated in the following tentative list:

The collection and disposal of refuse from an engineering standpoint and the utilization of municipal wastes; the handling of sewage sludge; the training of municipal employees; the cleaning of public buildings; the future of the police arm presented from the engineering side; the problem of organization as related to the highway department; controlling factors in municipal engineering; cleaning filter sands; the design and operation of a municipal electric-light plant; and municipal colleges in Germany. The session will be opened by John Purroy Mitchel, mayor of the City of New York. Arrangements for the session are being completed by the committee on public relations, and it promises to be one of the most attractive and largely attended meetings which the society has held.

The other feature is that of the reports which are expected for the Wednesday morning session. As is known to the membership, two of the most important reports ever undertaken by the society are nearing completion, and it is hoped that both can be presented at the coming annual meeting.

One report is that of the power test committee, which was published in preliminary form in the society's "Journal" nearly two years ago. Since then the report has been thoroughly revised and is now being put into its final form. It attempts to standardize the methods of testing the various types of prime movers and auxiliary apparatus, and is so broad in its scope that it will interest a large number of members. It is expected will constitute one of the most comprehensive publications that the society has ever issued.

The report of the committee on standardization of steam boilers is equally voluminous, and is also approaching its completed state. Its value both to engineers and the public at large is so great that it can scarcely be estimated, involving as it does every item of consequence in relation to the safety of steam boilers, from the specifications of steel to the qualifications of the men in the boiler room.

An interesting session, in charge of the subcommittee on iron and steel, will be devoted to subjects relating to cast iron and steel from the standpoint of the mechanical engi-

neer. New developments in the field of engineering metals will be dealt with in a way to bring out and make these recent advances a matter of common knowledge. Other sessions in charge of subcommittees will be those on machine-shop practice and on railroads. At the latter there will be a report by the subcommittee on railroads, on the steam locomotive of today. Copies of individual papers have been received from other subcommittees, and one session will be devoted to a number of miscellaneous papers of exceptional merit. The subjects will be announced later, when the complete program for the meeting will be published.

The president's reception will be given on Tuesday evening during the convention, and on Wednesday evening there will be a lecture upon a subject to be announced. The reunion this year is expected to take the form of a dinner dance at the Hotel Astor, and this will constitute the chief social event of the meeting. Covers will be laid for at least 350, and there will be dancing between courses. The grand ball room at the Astor has been engaged for this occasion.

Power on the Pacific Coast

In an address before the recent convention of the northwest Electric Light & Power Association at Spokane, Wash., W. E. Herring of the Puget Sound Traction Light & Power Co. showed that the Pacific Coast during the past few years has come forward with a greater increase in water power development than any other section of the United States.

The electrical energy as a whole is increasing at a tremendous rate. At the end of 1912 the total kilowatt rating of generators in the United States was in excess of 5,000,000, an increase of 323 per cent. in ten years. In no other industry has there been such an increase in capital, it now exceeding $7,000,000,000 and for the five-year period up to 1912 there was required an average of $180,000,000 a year for central station work alone.

Bringing the argument home to the Pacific Coast, Mr. Herring showed that the new capital needed on this slope in the next ten years would be from $150,000,000 to $250,000,000, depending to a great extent upon the number of new industries that can be brought to this section and upon the amount of electrification done by steam roads.

The minimum potential water power in the country shows a total of 26,736,000 hp., of which 4,932,000 is available in Washington, 3,148,000 in Oregon and 3,424,000 in California. The total for the three states is 11,504,000 hp, or nearly 43 per cent. of the total potential water power of the United States. Washington has developed 306,000 hp. or 6 per cent. of the possible power; Oregon 105,000 hp. or 3 per cent.; California, 430,000 or about 8 per cent.

With this wealth of power it is shown that in practically all of the Coast cities the value of the manufacturers' products is less than in a majority of Eastern cities of similar size; but it is believed that with a rapidly increasing population, the Panama Canal, and the opening of Alaska this condition will change.

Power Development in British Columbia

The Canadian Western Power Co. is making important improvements and increasing the capacity of its plant at Stave Falls, British Columbia. A 13,000-hp. turbine, a 900-hp. generator, a 3000 kw. transformer and a large outfit of switchboards will soon be installed for use in connection with the third unit of the plant, the buildings for which are nearing completion. The company has a capacity of 25,000 hp. Later, with the installation of the third unit, the plant will have 37,000 hp., and by next year will be generating over 50,000 hp. with four units in operation.

The new hydroelectric plant for Kamloops will get its power from the Barriere River. The power house building has been completed, the machinery is being erected and the city of Kamloops will be utilizing the current carried over the new line within a few weeks. The first unit will produce 2600 hp. and later the capacity of the plant will be increased to 8000 hp.

The city council of Revelstoke has awarded contracts for the improvement of the civic water power plant to Escher, Wyss & Co., Zurich, Switzerland, for a water turbine with a 44-in. valve, and to the Westinghouse Co. for a generator, exciter and switchboard. The total cost of the improvements will be about $20,000.

The Canadian Northern Ry. is developing a 50,000-hp. plant from the Sumallo and Nicaloon Rivers, which will be employed in operating the Canadian Northern trains from Port Mann to the False Creek terminals at Vancouver, B. C.

Announcement has just been made of the proposed construction of a large hydroelectric plant, to represent at its completion an expenditure of several millions, and to be on the Speel River, 35 miles south of Juneau, Alaska. The government has issued a grant for the projected undertakings and actual construction work is expected to start in a few months. The plant will have a capacity of 100,000 hp.

Salt Lake City to Investigate Smoke Nuisance

If Salt Lake City, Utah, will make the necessary appropriation, the metallurgical bureau of the Universary of Utah will undertake a systematic study of the smoke problem in Salt Lake.

A resolution recently adopted by the board of governors of the Commercial club was forwarded to the city commission asking that an appropriation to cover the cost of the investigation be made. The resolution was accompanied by a plan for a systematic investigation, evolved by Prof. D. A. Lyon, government expert in charge of the university bureau. In a letter to the city commission George W. Snow recommends that the officer of the United States Bureau of Mines at the University of Utah be allowed to conduct an investigation as to the smoke nuisance in Salt Lake, and that the city make an appropriation to cover the cost of the investigation.

Drought Shuts Down Plants

A large number of industrial plants in Pennsylvania, especially in the western section, have been compelled to shut down temporarily for lack of water. The long drought has dried up rivers and smaller streams upon which many companies depended for their water supplies. Some of the big coal companies are compelled to haul water long distances in order to keep their power plants in operation. West Virginia and Ohio report similar conditions. The drought is the worst that has visited the section in many years.

Conference of Mayors at Philadelphia

When it is remembered that over eight billion dollars are invested in electric light and street-railway companies alone in this country, five and one-half billions being controlled by holding companies, it will be realized what a stupendous task a single city has when, unaided, it asks for adequate rates and proper service from the public utilities.

This condition is largely responsible for the formation of the Utilities Bureau under the sponsorship of Mayors Blankenburg of Philadelphia, Mitchel of New York, Harrison of Chicago, Baker of Cleveland and Shroyer of Dayton. Its object is to secure fairer rates and better service standards from municipal utilities through the coöperation of public officials and interested citizens. The bureau is nonpartisan and at the service of those citizens and officials who need advice and seek data concerning other utilities than their own.

A conference of mayors will be held in Philadelphia on Nov. 12, 13 and 14. The program is as follows:

Reception Bellevue-Stratford Hotel, Thursday evening, Nov. 12. Address of welcome, Hon. John K. Tener, governor of Pennsylvania.

First Session—Bellevue-Stratford Hotel, Broad and Walnut Sts., ball room.

Friday Morning, 10:30 o'clock—"Practical Utility Problems." Presiding officer, Hon. Rudolph Blankenburg, mayor of Philadelphia. Addresses: "Fundamental Planks in a Public Utility Program," Delos F. Wilcox, franchise expert, New York; "The Regulation of Municipal Utilities," Nathaniel T. Guernsey, general counsel, American Telegraph & Telephone Co., New York; "Philadelphia's Transit Problem," A. Merritt Taylor, director of city transit, Philadelphia; "Municipal Lighting Rates," Ray Palmer, commissioner of gas and electricity, Chicago, Ill.; "Labor's Interests in Public Utilities."

Second Session—Bellevue-Stratford Hotel, Friday afternoon, Nov. 13, 2:15. "The Regulation of Utilities." Addresses: "Some Present Day Issues in Regulation," Edward W. Bemis, Chicago, Ill., member of board of supervising engineers, Chicago, valuation bureau of public utilities commission of the District of Columbia; "What Certain Cities Have Accomplished Without State Regulation," Stiles P. Jones, secretary, Voters League, Minneapolis, Minn.; "A Constructive Policy

for Public Service Corporations," Charles Day, of Day & Zimmerman, Philadelphia; "What Regulation Must Accomplish If It Is To Be Permanent," Hon. John M. Eshelman, president, railroad commission of California.

Third Session—Central High School Assembly Room. Under the auspices of the American Academy of Political and Social Science, Friday evening, Nov. 13, 8:15. "Local and State Regulation of Municipal Utilities." Presiding officer, Hon. John Purroy Mitchel, mayor of New York. Addresses and Discussion: Hon. Halford Erickson, member of railroad commission of Wisconsin; Hon. Milo R. Maltbie, member of public service commission for the first district of New York; Prof. Charles E. Merriam, professor of political science, University of Chicago; Hon. Ira W. Stratton, mayor of Reading, Penn.

Fourth Session—Bellevue-Stratford Hotel Ball Room, Saturday morning, Nov. 14, 10:30. "Municipal Ownership and Operation." Presiding officer, Hon. Newton D. Baker, mayor of Cleveland, Ohio. Address—"Municipal Ownership—The Testimony of Foreign Experience," Frederick C. Howe, commissioner of immigration, New York. Fifteen-minute reports from the managers of typical municipal plants. Open discussion led by Clarke M. Rosecrantz, of general counsel, Milwaukee Electric Ry. & Light Co., Hon. Robert Crosser, United States congressman from Ohio, member of committee on the District of Columbia.

Fifth Session—Saturday afternoon, Nov. 14, 2:15. Executive session, open only to mayors and delegates. Round table discussion: "Elements in a Constructive Utility Program."

Sixth Session—Witherspoon Hall. Under the auspices of the American Academy of Political and Social Science, Saturday evening, Nov. 14, 8:15. "Holding Companies and the Public Welfare." Presiding officer, Hon. Franklin K. Lane, secretary of the interior, Washington, D. C. Addresses and discussion: Francis T. Homer, Bertron, Griscom & Co., New York; Charles F. Mathewson, of counsel, Consolidated Gas Co., New York; James P. Goodrich, director, National City Bank, Indianapolis, Ind.; Louis D. Brandeis, lawyer, Boston, Mass. The attendance of 106 mayors or their delegates has been promised for the conference.

TRADE CATALOGS

American District Steam Co., North Tonawanda, N. Y. Catalog No. 9. Steam heating specialties. Illustrated, 234 pp., 6x9 in.

Sprague Electric Works, 527-31 W. 34th St., New York. Bulletin No. 48,701. Electric Dynamometers. Illustrated, 24 pp., 8x10½ in.

The Ohio Blower Co., 5130 Perkins Ave., Cleveland, Ohio. The HANDiLINE HANDiBOOK. Sheet metal and angle iron specialties including refuse boxes, cans, racks, battery cases, etc. Illustrated, 3½x6 in.

Ingersoll-Rand Co., 11 Broadway, New York. Form No. 4033. "Little Tugger" air hoist for power plants, etc. Illustrated, 8 pp., 6x9 in. Form No. 8013. "Little David" pneumatic chipping, calking and scaling hammers. Illustrated, 12 pp., 6x9 in. Form No. 8207. "Little David" pneumatic drills. Illustrated, 36 pp., 6x9 in.

NEW PUBLICATIONS

MINERS' CIRCULAR 16. Hints on coal-mine ventilation, by J. J. Rutledge. 1914. 22 pp.

TECHNICAL PAPER 65. A study of the oxidation of coal, by H. C. Porter and O. C. Ralston. 1914. 30 pp., 12 Figs.

NEW GOVERNMENT PUBLICATIONS—The Bureau of Mines, Department of the Interior, announces the following new publications:

TECHNICAL PAPER 93. Quarry accidents in the United States during the calendar year 1913, compiled by A. H. Fay. 1914. 66 pp.

TECHNICAL PAPER 84. Production of explosives in the United States during the calendar year 1913, by A. H. Fay. 1914. 13 pp.

BULLETIN 83. The humidity of mine air, with especial reference to coal mines in Illinois, by R. Y. Williams. 1914. 69 pp., 7 Figs.

NOTE—Only a limited supply of these publications is available for free distribution, and applicants are asked to cooperate in insuring an equitable distribution by selecting publications that are of especial interest. Requests for all papers cannot be granted without satisfactory reason. Publications should be ordered by number and title. Applications should be addressed to the Director of the Bureau of Mines, Washington, D. C.

SIMPLE PROBLEMS IN MARINE ENGINEERING DESIGN—By J. W. M. Sothern, Principal, Sothern Marine Engineering College, Glasgow. Third edition. James Munro & Co., Ltd., Glasgow; D. Van Nostrand Co., New York. 191 pages; 5x8-in. semi-flexible cloth. Price, $1.

This work contains data and methods of solving problems of design of boilers and engines, reduced to simple arithmetical and algebraical processes. The test questions under each head afford valuable information and practice for young engineers and candidates for licenses who may wish to refresh their knowledge of engineering calculations.

POCKETBOOK OF MARINE ENGINEERING RULES AND TABLES. By A. E. Seaton and H. M. Rounthwaite; 12th edition, with diagrams. London, Charles Griffin & Co.; New York., D. Van Nostrand Co. Flexible leather; 4x6½ in.; 713 pages. Price $3.50.

This pocketbook, which in Great Britain is known as a standard, is a compendium of rules, tables and data that should find a welcome place in the kit of every marine engineer. The rapid modifications of conditions in marine engineering of recent years, such as the adaptation of turbine and internal-combustion engines to propulsion of ships, have called for convenient form of additional data not included in earlier editions of this work. The authors have also included new Rules of the Board of Trade and the more recent modifications of Lloyds and other registry societies.

PUMPING BY COMPRESSED AIR—By Edmund M. Ivens, M. E. First edition. New York, John Wiley & Sons. 244 pages; 6x9 in.; 106 illustrations; cloth. Price, $3.

Literature on pumping by compressed air has heretofore been almost exclusively confined to air compressor catalogs, articles scattered through the proceedings of technical societies and some data to be found in works on compressed air which were either a repetition of material contained in catalogs or of a purely theoretical character.

The work will therefore be appreciated by engineers and others for its systematic and practical presentation by an author who has had the opportunity to install and test a number of air lifts operated under a wide range of conditions and is fortified by valuable data. The work places before the student a comprehensive theoretical study of the subject and at the same time imparts easily understood instruction to the operating engineer in the economic essentials of the practical installation.

In addition to the space devoted to design and operation of air lifts, the work contains good descriptions and illustrations of various commercial and return systems, and for greater completeness of the subject of compressed air pumping, the later chapters are devoted to pertinent principles of hydraulics and thermodynamics. The concluding chapters treat of air compressors, flow of compressed air in pipes and flow of water in pipes.

"VERBAL" NOTES AND SKETCHES, FOR MARINE ENGINEERS. By J. W. M. Sothern, Principal, Sothern's Marine Engineering College. Glasgow, James Munro & Co., Ltd.; New York, D. Van Nostrand Co. 676 pages. 6x9 in.; illustrated; cloth. Price, $5.

The former editions of this work are well known to the engineering profession, especially to those specializing in marine-engineering construction and operation. The work is systematically divided into ten sections and an appendix treating of workshop practice, in which are described such subjects as types of engines, design and construction of parts, auxiliary machinery, trial trips, etc.; marine-boiler construction and management; notes and sketches of various details; valves and valve data; general notes on designing and repairing of parts and marine-engineering operation; marine-engineering chemistry notes embracing practical chemistry of combustion, feed water and boiler and condenser corrosion; marine electric lighting; propellers; refrigeration, and internal-combustion engines; together with an appendix on marine steam turbines, air pumps and various details.

The work is to be especially commended for the thoroughness and completeness of treatment, good presentation and excellent illustrations, most of which are drawn to scale with the parts especially referred to emphasized with appropriate shading. The work is an excellent textbook for students and apprentice engineers and a valuable reference handbook on British marine engineering.

ENGINEERING AFFAIRS

Annual Meeting of Massachusetts Chapter, A. S. H. & V. E.—The annual meeting of the Massachusetts Chapter of the American Society of Heating and Ventilating Engineers was held Oct. 13, at the Revere House, Boston, Mass., with a large attendance. The principal business was the election of officers which resulted in the following: Andrew G. Paul,

president; Eugene R. Stone, vice-president; Charles Morrison, secretary; William T. Smallman, treasurer. Board of Governors: William G. Snow, Frank Irving Cooper, James W. H. Myrick. There is a good balance in the treasury. Several new members were admitted and a general feeling of enthusiasm and coöperation prevailed which bids fair to continue throughout the year. Resolutions were engrossed on the recent death of Albert Barnes Franklin, who was active in the society, having been second vice-president.

Pennsylvania Department of Labor and Industry—In accord with the unanimous vote of the first Pennsylvania Industrial Welfare and Efficiency Conference held in Harrisburg, Penn., last year, Commissioner Jackson has issued a call for a second conference to be held in the State Capitol at Harrisburg on Nov. 17, 18 and 19. This conference is held under the auspices of the Pennsylvania department of labor and industry and the Engineers Society of Pennsylvania. The purpose is to enable the employers and employees to work out together the great problems before them with reference to increasing the welfare of the employees and the prosperity of the industries. The conference last year was attended by some 2000 persons, many of whom were leaders in the labor and industrial world. The gathering this year bids fair to have an even larger attendance and to create even more interest than that of last year.

In connection with the conference proper, will be held an unusually effective safety, welfare and efficiency exhibition which will open on Nov. 16 and close on Nov. 20.

The commissioner believes, both from the unusual value of the conference last year and the even more positive purposes of the present conference, that a great amount of work can be accomplished this year. He therefore earnestly calls upon every broad-minded citizen who is interested in industrial affairs to come to Harrisburg at the time of the conference and take an active part in the proceedings.

PERSONALS

Paul Bigelow, Eastern manager of the Buckeye Engine Co., addressed the Brooklyn Engineers' Club, 117 Remsen street, Brooklyn, N. Y., on the evening of Oct. 29 on the Buckeye-mobile.

George H. Chandler, a stationary engineer for more than 50 years, and for the past 17 years the chief engineer of the Bryant Paper Co., Kalamazoo, Mich., has retired from that position at the age of 74 years. Mr. Chandler will act as storekeeper for the company, and in recognition of his long and faithful service will draw the same salary he had while chief engineer.

Dr. John A. Brashear, Pittsburgh, Penn., has been nominated for the presidency of the American Society of Mechanical Engineers. Other nominations are as follows: For vice-presidents: Henry Hess, Hess Steel Castings Co., Bridgeton, N. J.; G. W. Dickie, San Francisco, and James E. Sague, Public Service Commission of the Second District, New York. For managers: Charles T. Main, Dean & Main, Boston; Spencer Miller, Lidgerwood Manufacturing Co., New York; Max Toltz, St. Paul, and Morris L. Cooke, director of public works, Philadelphia. William H. Wiley, John Wiley & Sons, New York, has been renominated as treasurer.

George M. Brill has been appointed director of the general reception committee for the International Engineering Congress to be held in connection with the Panama-Pacific Exposition in San Francisco next year. With headquarters in the Engineering Societies Building in New York City he is to organize reception committees in the large cities of the country. Among the members of the general reception committee are Walter M. McFarland, Babcock & Wilcox Co.; William L. Saunders, Ingersoll-Rand Co., New York; E. D. Meier, Heine Steam Boiler Co.; Ambrose Swasey, Warner & Swasey Co., Cleveland; Howel H. Barnes, Jr., General Electric Co.; Stevenson Taylor, Quintard Iron Works, and Dr. A. C. Humphreys, Stevens Institute of Technology.

OBITUARY

CURTIS E. WHITTLESEY

Curtis E. Whittlesey, for many years treasurer and secretary of the McGraw Publishing Co., died at his home in Madison, N. J., on Oct. 9, in his 72d year.

During the 24 years Mr. Whittlesey spent in the publishing business he gained for himself a host of friends who honored and loved him for his gentleness and his unostentatious deeds of real benevolence. His knowledge of finance made him frequently sought after by business men who valued his discernment and business acumen very highly.

JOSHUA H. MILLETT

Joshua Howard Millett, president of the Crosby Steam Gage & Valve Co., died at his home, Malden, Mass., on Oct. 15, of hardening of the arteries, aged 72 years, 6 months, 28 days. He had been ill but a few days and was at his office in Boston within three weeks.

Mr. Millett was born in Cherryfield, Me., and was graduated from Colby University and was later admitted to the bar. For many years he practiced successfully in Boston, but retired from active practice in 1895 to devote his time to the concern of which he was the head. When admitted to the bar he became a member of the firm of Redfield, Herrick & Millett, of which Judge Isaac Redfield was the head.

He came to Malden in 1869 and was soon identified with the welfare of the town. He was elected to the House in 1884 and 1885 and served on the committee on judiciary and the special committee on establishing the metropolitan police. He was a member of the special commission to revise the city charter and served on the park commission and school committee and was for a time a trustee of the public library. He also served on the committee to erect the soldiers monument. He was urged several times to take the mayoralty and to stand for the senate and for congress but in later years refused all political honors.

He was a member of Converse lodge of Masons, Tabernacle chapter, Melrose council and Beauseant Commandery, the Historical society, the Sons of the American Revolution and the college Greek letter fraternity of D. K. E. For many years Mr. Millett was Malden's leading Unitarian layman and was much interested in the Home for Aged Persons and was president of the corporation from its incipiency until his death. Mr. Millett was a patron of art to a considerable extent and many fine paintings hung in his beautiful home.

An officer of the Crosby company who was closely associated with Mr. Millett has this to say of him: "In my own varied business experience, I have come into close contact with a great many men, of many types; but I have never known another of his like, and his many personal and business acquaintances have used these same words in regard to him during all the past 10 years that I have known him intimately. This is merely to give an idea of how he was regarded by those who could best estimate him, and possibly to throw a side light upon the controlling policy which has established the Crosby company upon a high plane of reputation and accomplishment."

Besides his widow he leaves a son, Charles H. Millett, associated with him in business and a daughter, Mrs. Alfred B. Carhart of Winchester.

Massachusetts Board of Boiler Rules Hearing

On Thursday, Oct. 15, the Massachusetts Board of Boiler Rules held the hearing prescribed by the statute under which it is appointed, and discussed the proposed changes which were published on page 395 of our issue of Sept. 15. The hearing was attended by about one hundred engineers, and the remonstrance against making any change in the rules in advance of the publication of the report of the committee on uniform boiler specifications of the American Society of Mechanical Engineers, was practically unanimous, the only interests appearing to favor the holding aloof by Massachusetts from the nation-wide effort to obtain uniform laws appearing to be the boiler manufacturing interests of that state.

The proposed change which occasioned the most discussion was that reducing water-tube boilers with cast headers to 15 lb. pressure, which would put millions of dollars worth of water-tube boilers already installed out of service. It was shown that owing to improved methods of manufacture, the cast header is being gradually abandoned, and that the history of its use does not show it to be so serious a menace as to warrant such drastic legislation.

There are at least three, and perhaps four, courses open to the board; one to reject the proposed changes in toto; the other to adopt them in toto; the third to revise them in accordance with the light evolved by the hearing, and holding a new hearing after the advertisement required by law; or perhaps to adopt such of the changes as commend themselves to the board after the discussion. If any attention is to be paid to the sentiments expressed at the hearing, nothing will be done until the final report of the American Society of Mechanical Engineers' committee is submitted to the December meeting of that society.